THE EFFECT OF ENERGY RECOVERY ON INDOOR CLIMATE, AIR QUALITY AND ENERGY CONSUMPTION USING COMPUTER SIMULATIONS

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ABSTRACT

The main objectives of this thesis are to determine if the addition of an energy wheel in an HVAC system can improve the indoor air relative humidity (RH), and perceived air quality (PAQ), as well as reduce energy consumption. An energy wheel is an air-to-air energy exchanger that transfers heat and moisture between the outdoor air entering and the exhaust air leaving a building. This thesis uses the TRNSYS computer package to model two buildings (an office and a school) in four different cities (Saskatoon, Saskatchewan; Vancouver, British Columbia; Tampa, Florida and Phoenix, Arizona).

The results with and without an energy wheel are compared to see if the energy wheel has a significant impact on the RH and PAQ in the buildings. The energy wheel reduces peak RH levels in Tampa, (up to 15% RH), which is a humid climate, but has a smaller effect on the indoor RH in Saskatoon (up to 4% RH) and Phoenix (up to 11% RH), which are dry climates. The energy wheel also reduces the number of people that are dissatisfied with the PAQ within the space by up to 17% in Tampa.

The addition of the energy wheel to the HVAC system creates a reduction in the total energy consumed by the HVAC system in Saskatoon, Phoenix and Tampa (2% in each city). There is a significant reduction in the size of the heating equipment in Saskatoon (26%) and in the size of the cooling equipment in Phoenix (18%) and Tampa (17%). A cost analysis shows that the HVAC system including an energy wheel has the least life-cycle costs in these three cities, with savings of up to 6%. In Vancouver, the energy wheel has a negligible impact on the indoor RH, PAQ and energy consumption.

ACKNOWLEDGEMENTS

I would like to express my sincere thanks to my supervisor C.J. Simonson for his advice and guidance in this research and in the preparation of this thesis. I would also like to convey special thanks to my husband Ryan for his support and encouragement during this research.

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NOMENCLATURE

ACRONYMS:

ach	air changes per hour
ASHRAE	American Society of Heating, Refrigeration and Air-Conditioning Engineers
C	Cooling unit
E	economizer
EA	exhaust air
EF	exhaust fan
H	heater
HVAC	Heating, Ventilating and Air Conditioning
IAQ	indoor air quality
OA	outdoor air
PAQ	perceived air quality
RA	recirculation air
SA	supply air
SF	supply fan

SYMBOLS:

A	floor area [m^2] (used in calculating ventilation rates)
Acc	acceptability of air
C_{aux}	cost of auxiliary heating and cooling power [$/kWh]
$C_{aux,N}$	cost of auxiliary heating and cooling equipment without an energy wheel [$/kW]

$C_{aux,ex}$	cost of auxiliary heating and cooling equipment with an energy wheel [$/kW]
C_{eq}	cost of auxiliary heating and cooling equipment [$/kW]
C_{ex}	cost of energy wheel [$/(L/s)]
C_{fan}	cost of fan power [$/kWh]
$C_{p,air}$	constant-pressure specific heat of air [J/(kg·K)]
C_{rec}	savings due to heating and cooling recovered by the energy wheel [$/kW]
Cr_o^*	overall matrix heat (or moisture) capacity ratio
Crm_o^*	overall matrix moisture capacity ratio
C_{tot}	life-cycle cost [$]
clo	unit of clothing
e_c	cost of electricity [$/kWh]
e_h	cost of natural gas [$/kWh]
h_{fg}	latent heat of vaporization of air [J/kg]
H	enthalpy of air [kJ/kg]
H*	operating condition factor that represents the ratio of latent to sensible enthalpy differences between the supply and exhaust air inlets of the energy wheel
LCC	life-cycle cost [$]
$\dot{m}$	mass flow rate of air [kg/s]
met	metabolic rate per unit of body surface area
NTU_o	overall number of transfer units
P	number of people (used in calculating ventilation rates)
P_{wef}	present worth escalation factor
PB	payback period [years]

PD	percent dissatisfied [%]
Q_l	latent heat transfer [W]
Q_s	sensible heat transfer [W]
Q_{vent}	total outdoor air flow rate [L/s]
$q_{tot,c}$	total electricity required for cooling [kWh]
$q_{tot,fan}$	total electricity required for fan power [kWh]
$q_{tot,h}$	total energy required for heating [kWh]
R	thermal resistance [$(m^2 \cdot K)/W$] or [$(h \cdot ft^2 \cdot F)/BTU$]
R_p	occupant factor (used in calculating ventilation rates)
R_a	area factor (used in calculating ventilation rates)
RH	relative humidity [%]
T	temperature [K or °C]
u	mass fraction of water in the desiccant [kg_w/kg_d]
U-value	overall heat transfer coefficient [$W/(m^2 \cdot K)$] or [$BTU/(h \cdot ft^2 \cdot F)$]
W	humidity ratio [g_w/kg_d] or [kg_w/kg_d]
Wm	empirical coefficient used in the sorption isotherm describing the maximum moisture capacity of the desiccant [kg_w/kg_d]
X	fraction of recirculation air used

Greek Symbols:

Δ	difference between two values
ε_l	latent heat transfer (or moisture transfer) effectiveness
ε_s	sensible heat transfer effectiveness

η	fraction of the phase change energy that is delivered directly to the air
ϕ	relative humidity

Subscripts:

ave	average value
base	results from the base case simulations
bypass	economizer bypass value above which the bypass economizer can be used to meet the building space loads
constant	results from the constant effectiveness energy wheel simulations
critical	critical value above which the energy wheel is operating at part load conditions
design	design value
e,i	exhaust side inlet conditions of the energy wheel
energywheel	results from the constant effectiveness energy wheel simulations
in	indoor air conditions
max	maximum value
min	minimum value
o	initial value
outdoor	outdoor air conditions
s	supply air conditions
s,i	supply side inlet conditions of the energy wheel
tc	thermal comfort
variable	results from the variable effectiveness energy wheel simulations

CHAPTER 1

INTRODUCTION

One of the most important goals when designing heating, ventilating and air conditioning (HVAC) equipment for a building is to satisfy the people that occupy that building. Although it is impossible to design a building so that every occupant is comfortable, it is important to ensure that the majority of occupants are comfortable. There are several factors that have been shown to affect the comfort or satisfaction of occupants, including: temperature, relative humidity, air velocity, contaminants, odors, dust, noise, lighting and vibrations. This thesis will examine the effect of temperature and relative humidity on thermal comfort, indoor air quality and perceived air quality. Computer simulations are performed with and without an energy wheel to determine the effect of air-to-air energy recovery on indoor relative humidity levels as well as energy consumption in a building. The computer simulations are performed using a computer simulation program called TRNSYS (Solar Energy Laboratory, 2005).

1.1 THERMAL COMFORT

Thermal comfort can be divided into two categories: general thermal comfort and local thermal comfort. General thermal comfort refers to the overall comfort level of a person. Local thermal comfort refers to the comfort level of a specific part of a person, for example, sitting beside a cold window could cause one side of a person to be cold.

In most commercial and residential buildings, thermal comfort is dominated by the temperature of the air within the space. If the temperature is too high or too low the occupants will not be comfortable. According to ASHRAE Standard 55 - Thermal Environment Conditions for Human Occupancy (2004), there are other factors that can also contribute to the comfort levels of the occupants. These factors are the metabolic rate of the occupants, amount of clothing worn by the occupants, and the radiation, air speed and relative humidity levels in the space.

The human body produces heat through metabolic activities which must be dissipated to the surroundings in order to maintain normal body temperatures. This metabolic activity is characterized in terms of heat production per unit area of skin. A person sitting at rest produces approximately 58 W/m^2 (18.4 $BTU/(hr \cdot ft^2)$) of heat, based on a skin surface area of 1.8 m^2 (19.6 ft^2). This amount of heat is called 1 met. People that are engaging in more strenuous activities and therefore have higher metabolic rates produce more heat. A person working at a metabolic rate 5 times the resting rate would have a metabolic rate of 5 met. Due to the increase in the amount of heat being produced, a person with a high metabolic rate will feel uncomfortable at an air temperature that would feel comfortable to a person generating a lower amount of heat. No matter what the activity level of an occupant is, the clothing that they are wearing will act as an insulator. If a person is wearing a large amount of clothing (e.g. several layers of thick fabric) he or she will feel much hotter than a person wearing a small amount of clothing (e.g. shorts and a t-shirt). This factor must also be taken into account when designing the space conditions in buildings.

The final three factors mentioned above, that will affect thermal comfort (radiation, air speed and relative humidity) are environmental factors. A person gains or loses heat by radiation heat transfer with the surrounding surfaces. The surfaces in a room may be hotter (e.g. a stove or fireplace) or colder (e.g. a window or wall in a cold climate) than the actual air temperature in the room. When a person is exposed to these surfaces the person may feel hotter or colder than would be expected based on the air temperature alone. The impact of the radiation on thermal comfort depends on the temperature of the surface, view factors and the convection heat transfer between the person and the room air. If the temperature of the surface is similar to the air temperature then radiation will not have a significant effect. If the surface temperatures are quite different from the air temperature then radiation will have a larger impact on the thermal comfort of the person.

The air speed in a space also affects thermal comfort. This usually pertains to the air entering the space through vents, but it can also include air entering through infiltration, or through open windows or doors. High air speeds cause high convection coefficients between the body and the surrounding air. This causes more heat to be removed from the body making the person feel cooler. For a given air temperature, occupants that are more active or wearing more clothing may find the high air speeds acceptable, whereas the occupants that are less active and have less clothing may not. The high air speeds may make the less active people feel too cool unless the air temperature is increased. Conversely, very low air speeds result in a low convection transfer coefficient and therefore the occupants may feel hotter than the actual temperature of the air might indicate.

The remaining factor that can affect thermal comfort is the humidity level in the space. Low indoor humidity levels can cause the space to feel too dry and people may experience physical discomfort such as dry and itchy eyes and skin. If the indoor temperature and humidity levels are high, the air temperature can feel hotter than it actually is. When the temperature is high, people are naturally cooled by the perspiration evaporating from their bodies. If the humidity level is high, the moisture evaporates at a much slower rate than it would at a low humidity level and therefore the evaporative cooling of the body is lower at high humidity levels.

The temperature and humidity of the indoor air can also cause local thermal discomfort to a person in the form of warm respiratory discomfort. The respiratory tract regulates the temperature and humidity content of inhaled air on its way to the lungs. The inhaled air causes cooling of the mucous membranes in the upper respiratory tract. This cooling occurs in two manners, convective and evaporative heat loss. A low air temperature will cause a large amount of convective cooling and air at low humidity levels causes a large amount of evaporative cooling. For high temperature and humidity levels, the amount of cooling decreases and the air may be perceived as stuffy and uncomfortable.

Toftum et al. (1998a and b) determined upper limits for indoor air humidity to avoid uncomfortably humid skin and to prevent warm respiratory discomfort. The first study showed that the relative air humidity could increase to 100% RH with only a moderate number of people being dissatisfied due to humid skin, In the second study, the temperature and humidity were shown to have an impact on the perception of respiratory thermal sensation, freshness and acceptability. When the convective and evaporative cooling was high, the people felt that the air was cool. As the amount of cooling decreased, the people perceived the air to be increasingly stuffy. The acceptability of the air was higher when the people felt the air was cool and fresh, as opposed to warm and stuffy. The high acceptability ratings occurred at low temperature and humidity levels. A model to predict the percent dissatisfied with warm respiratory comfort was created.

According to ISO Standard 7730 (1994), the relative humidity in a space should be between 30% RH and 70% RH to decrease the risk of wet or dry skin, eye irritation, respiratory diseases and microbial growth. ASHRAE Standard 55 (2004) gives the upper limit of humidity for thermal comfort as 0.012 kg/kg. According to Standard 55 there are no established lower limits, but there are several factors, including skin drying, irritation of mucous membranes and dryness of eyes that may put limits on the acceptability of low humidity levels. The percentage of people dissatisfied with thermal comfort (PD_{tc}) is calculated by the computer program based on ISO Standard 7730.

1.2 INDOOR AIR QUALITY (IAQ)

Indoor air quality (IAQ) is a measurement of the concentration of contaminants in the air. There are numerous gaseous contaminants (e.g. ASHRAE Fundamentals (2005) lists 40) such as volatile organic compounds, carbon dioxide, and carbon monoxide, as well as microorganisms, viruses and allergens. The most common detection of poor air quality is through odor. To maintain good IAQ, these contaminants must be kept below certain levels. Contaminants can be produced in the space by the occupants breathing or by other processes within the space (e.g. cooking). They can be emitted from furniture within the space, or introduced into the room through infiltration and ventilation using contaminated outdoor air. Poor IAQ can cause the occupants to feel sick which can result in a loss of productivity. Several studies have been done by Kosonen and Tan (2004a and b) showing that a loss of productivity can result from an increase in temperature and from the occupants' perception of the indoor climate. A commonly practiced method of maintaining adequate IAQ in commercial and residential buildings is the use of outdoor ventilation.

Perceived Air Quality (PAQ) is a measure of how the occupants in a space experience or perceive the quality of the indoor air conditions. Even if the actual IAQ is good, that is, the air is free of contaminants; the air can seem to be of poor quality if the temperature and humidity of the air are too high. If the PAQ is poor, then the percentage of people dissatisfied with the space will be high. When designing buildings, it is important to ensure that a building has not only good IAQ but also good temperature and humidity levels for the occupants. Studies by Fang et al. (1998a and 1998b) have shown that temperature and humidity have a strong impact on the perceived air quality of a space, and that as the temperature and humidity increase beyond a comfortable level, the perceived air quality decreases. They found a linear relationship between the acceptability of air and the enthalpy of the air. One study was performed for an initial reaction to the air conditions and the other a whole-body exposure for 20 minutes. It was found that the acceptability does not change during the 20 min period, but is the same as the first impression. The model they generated is used in this thesis to calculate the acceptability of air and subsequently the percent dissatisfied with PAQ (PD_{IAQ}). The equation for the acceptability of air is

$$\mathrm{Acc} = -0.033\mathrm{H} + 1.662, \tag{1.1}$$

for clean air, with H = enthalpy of the air [kJ/kg]. The percentage of people dissatisfied is then calculated by

$$PD_{IAQ} = \frac{\exp(-0.18 - 5.28Acc)}{1 + \exp(-0.18 - 5.28Acc)} * 100. \tag{1.2}$$

With equations (1.1) and (1.2), the effect of air temperature and humidity on PD_{IAQ} can be estimated.

1.3 MODERATING RELATIVE HUMIDITY

One focus of this thesis will be on maintaining good PAQ by moderating humidity levels in buildings. There are currently several methods for moderating humidity levels in buildings. The two main methods of control include using outdoor ventilation and mechanical cooling equipment. Outdoor ventilation simply entails bringing fresh air from outdoors into the space. If the outdoor air is more humid than the space air, moisture will be added to the space air, and if the outdoor air is less humid than the space air, moisture will be removed. The problem with this method is that if the indoor air is too humid (above the upper humidity limits), and the outdoor air is also very humid there will be little or no moisture removal. For this reason, this method only works well in cold, dry climates. The other common method uses mechanical cooling equipment to cool and dehumidify the supply air being delivered to the space. The supply air will then be able to remove moisture from the space. This solves the problem of only being able to use outdoor air in cold, dry climates. This method however, has very large operational costs, including initial costs of purchasing and installing equipment and energy costs to run the equipment. Since neither of these methods is optimal, recent research has suggested another method: using hygroscopic materials in the space to moderate humidity levels.

A hygroscopic material is one that can readily absorb moisture. Wood is an example of a natural hygroscopic material. In a building, the hygroscopic material can be present in the walls and ceiling of a room, or in the furniture. If the air in a space is humid, the hygroscopic materials will absorb and store some of the moisture, reducing the humidity level in the space. If that space later on becomes too dry, the hygroscopic material will release the moisture back into the space increasing the humidity level of the space. In this manner, hygroscopic materials help to control the humidity level within a space. Simonson et al. (2002 and 2004) showed that the peak indoor humidity can be reduced significantly when using hygroscopic materials, as compared to non-hygroscopic materials for a set outdoor ventilation rate. They also found that it is possible to improve comfort and PAQ with the addition of hygroscopic materials to a room. They found that on average 2% more people will be satisfied with warm respiratory comfort and 6% more people will be satisfied with the PAQ in a room with hygroscopic materials than in a room with no hygroscopic materials.

Since using hygroscopic materials within a space can help moderate humidity levels and improve PAQ, this project was performed to determine if similar results can be achieved by moving the hygroscopic materials from the space and into the HVAC system, in the form of a desiccant coated energy wheel.

1.4 ENERGY RECOVERY

With the large amount of energy required to run a modern day building, it is important to design the HVAC system in a building to be as energy efficient as possible. This means reducing the amount of energy that typical buildings use. Saving energy is not only important for the environment, it also reduces the cost of running large buildings, by reducing the amount of electricity or other energy sources that the building consumes. One way of realizing these energy savings is by using air-to-air energy recovery devices. An energy recovery device transfers sensible and/or latent heat between the exhaust and supply air streams of a building. Although there are several different types of energy recovery devices this project utilizes a rotary air-to-air energy exchanger that can transfer both sensible and latent heat.

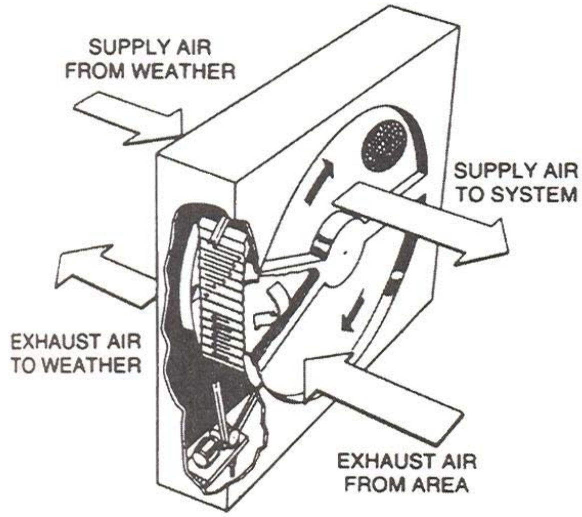


Figure 1.1. A schematic of an air-to-air energy wheel. (ASHRAE HVAC Systems and Equipment, 2004).

As shown in Figure 1.1, the air being supplied to the space passes through the top section of the wheel, from left to right, while the air being exhausted from the building passes through the bottom section of the wheel, passing from right to left. Inside the energy wheel, the air comes in contact with a surface which is coated with a desiccant material, that is, a material that absorbs water.

According to Jeong and Mumma (2005) the surface of most energy wheels is made of an aluminum matrix, with corrugated flow channels. The aluminum is in thin sheets (~0.07mm thick) and is corrugated to make tiny sinusoidal flutes. The majority of the sensible heat transfer is accomplished through this aluminum matrix. The matrix can also be made of non-metallic materials such as paper, plastic or synthetic fiber, but these are more commonly used when little sensible heat transfer is required. The most commonly used desiccant materials are silica gel and 3-4 Ǻ molecular sieve. Each desiccant has its own advantages and disadvantages. Silica gel can absorb high amounts of water and can stand up to acidic environments. It has good adsorption characteristics over a wide range of relative humidity but increasing temperature causes the adsorption capacity to decrease.

For sensible heat transfer the medium picks up heat from the incoming hot air stream and stores it. The wheel then rotates so that this section comes in contact with the cold air stream. The heat is released into the cold air stream, causing it to warm up. For latent heat transfer, the medium absorbs moisture from the more humid air stream. The wheel rotates to the other stream and the moisture is released through desorption into the less humid air stream. For total heat recovery both processes take place simultaneously.

The performance of an energy exchanger is described by its effectiveness to transfer sensible, latent and total energy. The effectiveness is the ratio of actual transfer for a given device to the maximum possible transfer between airstreams. According to Sauer et al. (2001) the average effectiveness for sensible and total heat exchangers is between 70 and 85% for equal supply and exhaust air mass flow rates. It is known that the effectiveness of a sensible heat exchanger is relatively constant regardless of small fluctuations in the inlet fluid temperature. According to Simonson et al. (1999a and b) however, the effectiveness of an energy wheel changes as the inlet humidity and temperature changes. As the supply inlet temperature and humidity increase, the latent effectiveness tends to decrease. The effectiveness of an energy wheel also depends on the slope of the sorption isotherm of the desiccant material used, as well as many design factors (face velocity, heat and mass transfer surface area, wheel speed, etc.). These parameters will be fixed for a given design, but the effectiveness will still change as the outdoor temperature and relative humidity change, therefore the effectiveness will change during the year. The impact of this on energy consumption has not been studied so it will be investigated in this thesis.

Since the effectiveness of an energy wheel depends on the outdoor temperature and relative humidity, the energy wheel simulations will be run under two conditions, with a constant effectiveness and with a variable effectiveness. In the first energy wheel case the effectiveness is assumed to be a constant throughout the year. In the second energy wheel case, the effectiveness will be determined using correlations from Simonson et al. (1999b) and will be allowed to vary with the outdoor temperature and relative humidity. The energy wheel is rated using ANSI/ARI Standard 1060 (2001) with the cooling season specifications.

1.5 OBJECTIVES AND METHODOLOGY

There are two main objectives in this research project. The first objective is to determine whether the relative humidity levels in a building can be moderated to decrease PD_{IAQ} using an energy wheel, as compared to a case with no energy wheel. The second objective is to determine whether any energy and cost savings can be realized with the use of an energy wheel. Within these two main objectives there are some smaller objectives as listed below.

- Determine the effect of an energy wheel on indoor relative humidity levels, PAQ, thermal comfort and energy savings.
- Determine the PD_{IAQ} for each space.
- Perform an economic analysis for each building in each city.
- Perform a sensitivity study to determine the effect of a new ventilation standard (ASHRAE Standard 62-2004), a lower recirculation rate, different energy wheel effectivenesses and the construction of the building on the simulation results.

The tasks that must be performed to accomplish these objectives are:

- Search for and select a commercial computer simulation program that includes total energy recovery and moisture storage in building materials as well as calculates the indoor temperature and relative humidity in a building.
- Using this program create a model of an office building and a school building in four different cities.
- Run simulations of the HVAC system for each of these buildings and output the temperature, relative humidity and comfort levels in each space as well as the energy consumption by the HVAC system.

CHAPTER 2

BUILDING DESCRIPTIONS

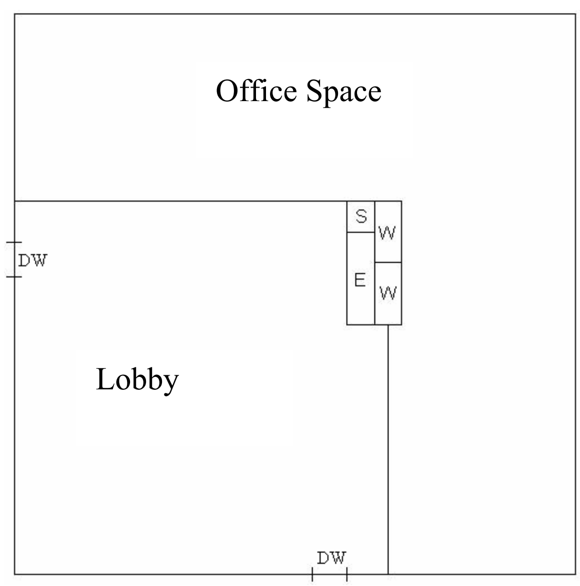
To determine if an energy wheel can improve relative humidity levels and reduce energy consumption in a building, a computer simulation is set up using the TRNSYS simulation package for two different buildings: a large office building and a school. A description of these buildings, including floor plan, construction, infiltration, ventilation and internal loads from occupants, lighting and equipment are given in this chapter. An example of the building file used by the program is found in Appendix A.

2.1 THE OFFICE BUILDING

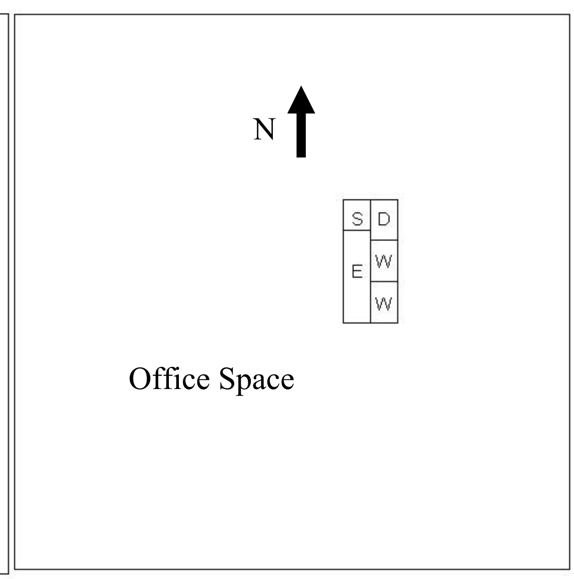
The office building that is used in this simulation is modeled after a building used by Dhital (1994), who tested the effects of a run-around heat exchanger on energy use and life cycle costs. A few modifications are made to the building to make it easier to input into the TRNSYS simulation program.

2.1.1 Floor Plan of the Office Building

The office building has 15 floors, 14 of which are occupied and therefore need to be conditioned. The top floor is used for mechanical purposes and is unoccupied so it is not conditioned. The building has dimensions of 40.5m x 40.5m (133ft x 133ft) and each floor is 2.7m (9ft) high. There is a 1m (3.3ft) high plenum located between each of the floors 1 through 14. The first floor consists mainly of a lobby and office spaces, while the remaining 13 floors are all office spaces, as seen in Figure 2.1 below. The lobby has two doors (DW), one facing south and one facing west, through which the people enter and exit the building. There is an elevator shaft (E) which contains three elevators and runs from the first floor to the 14th floor. The stairwell (S) runs all the way from the first floor to the 15th floor. A duct chase (D) starts on the second floor and continues up to the 15th floor. There are two washrooms (W) located on each of floors 1 through 14. Permanent walls exist around the stairwell, elevators, washrooms and duct chase only. The offices are all divided using moveable partitions. The office space, lobby, washrooms, elevator and stairwells are all supplied with outdoor air. Each of these spaces is considered a thermal zone, so that all of the offices are in one thermal zone, termed the office thermal zone, all of the washrooms are in the washroom thermal zone and so on.



(a)

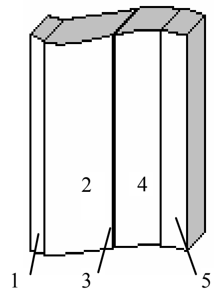


(b)

Figure 2.1. Floor Plan of the 1st floor (a) and 2-14th floors (b) of the office building.

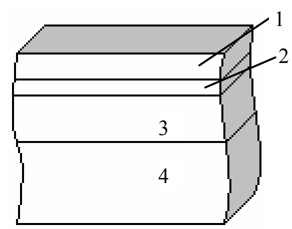
2.1.2 Construction of the Office Building

The exterior walls of the office building are made from 0.6 cm (0.24 in) of Architectural glass on the outside, 7.5 cm (2.95 in) expanded polystyrene insulation, 1 cm (0.39 in) steel back panel, a 5 cm (1.97 in) air space and 2 cm (0.79 in) gypsum board on the interior, as shown in Figure 2.2. The U value for this wall is 0.211 W/(m^2·K) (R = 26.9 h·ft^2·F/BTU). The building roof consists of 2.5 cm (0.98 in) gravel exterior, 1 cm (0.39 in) of built up roofing, 5 cm (1.97 in) of expanded polystyrene insulation and 10 cm (3.94 in) of concrete on the inside, as shown in Figure 2.3. The U value for the roof is 0.496 W/(m^2·K) (R = 11.5 h·ft^2·F/BTU). The windows are double glazed and have a U value of 2.7 W/(m^2·K) (R = 2.1 h·ft^2·F/BTU). They cover 40% of the exterior office wall area (27% of the exterior building area). The interior walls are made of 15 cm (5.91 in) of concrete. The ceiling between the space and the plenum above it is made of 2 cm (0.79 in) acoustic tile. The ground floor construction is 20 cm (7.87 in) of concrete with 5 cm (1.97 in) of polystyrene insulation below the concrete. The floor for each of the upper levels is made of 10 cm (3.94 in) of concrete. The thermal capacitance of the office and lobby are set at 12 times the space volume and the rest of the spaces in the building are set at 1.2 times the space volume. The office and lobby capacitance is set higher to account for the furniture in the spaces. The moisture storage of the materials is set at the default value given by the program. To perform a study of the moisture storage of the materials this parameter could be changed, but in this thesis it is left alone.



Material	Thickness (cm)
1. Architectural glass exterior	0.6
2. Expanded polystyrene insulation	7.5
3. Steel back panel	1
4. Air space	5
5. Gypsum board interior	2

Figure 2.2. Cross-section of the exterior walls of the office building (U = 0.211 W/(m^2·K), R = 26.9 h·ft^2·F/BTU).



Material	Thickness(cm)
1. Gravel (top)	2.5
2. Built-up roofing	1
3. Expanded polystyrene insulation	5
4. Concrete (bottom)	10

Figure 2.3. Cross-section of the roof of the office building (U = 0.496 W/(m^2·K), R = 11.5 h·ft^2·F/BTU).

2.1.3 Infiltration and Ventilation in the Office Building

The infiltration rate through the exterior walls of the building is 800 cm^3/(s·m^2) (0.157 cfm/ft^2), or 0.29 air changes per hour (ach) as found in Dhital (1994), which is based on information from McQuiston and Spitler (1992). This infiltration occurs 24 hours a day. The lobby has an additional infiltration rate of 8100 cm^3/(s·m^2) (1.59 cfm/ft^2) or 0.15 ach to account for the opening and closing of the doors. Since the doors will be open more in the morning, at lunch time and in the evening when the majority of people enter or exit the building, the lobby infiltration is scheduled as shown in Figure 2.4.

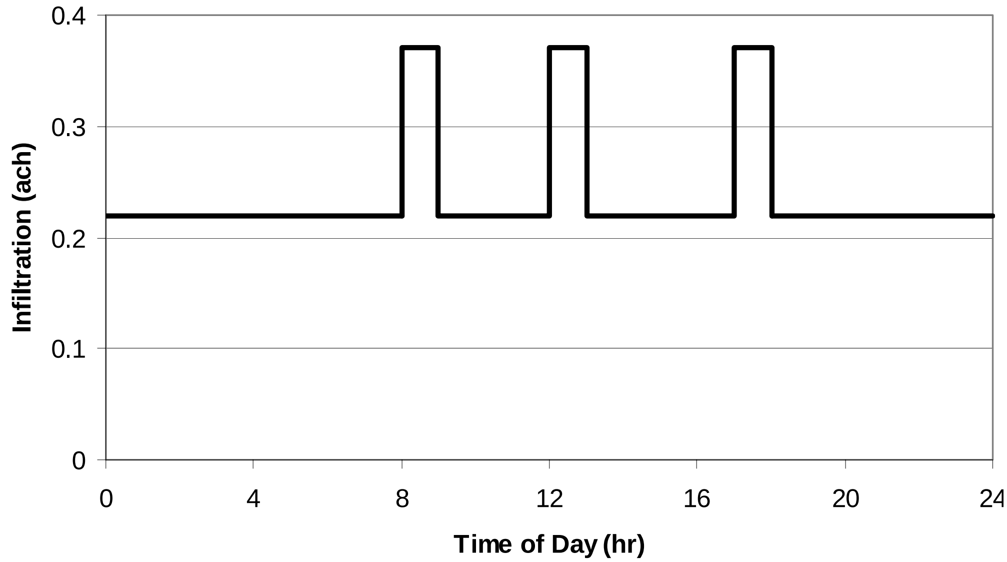


Figure 2.4. Infiltration schedule for the lobby in the office building.

The ventilation rates for each space in the office building are from ASHRAE Standard 62 (2001) Ventilation for Acceptable Indoor Air Quality. The total outdoor ventilation rate supplied to the building is 15,800 L/s (33,500 cfm) when the ventilation system is on. The ventilation system is turned on two hours prior to people entering the building (i.e., 6:00) and shut off two hours after the last occupants leave (i.e., 23:00). The ventilation rates for each space are given in Table 2.1.

Table 2.1. Ventilation rates for each zone in the office building.

Zone	**Ventilation Rate**
Office	10 L/(s·person)
Lobby	8 L/(s·person)
Washroom	25 L/(s·person)
Elevator	5 L/(s·m^2)
Stairs	0.50 L/(s·m^2)

Since the work on this thesis began, a new version of Standard 62 has been published, ASHRAE Standard 62.1 (2004). In the new standard the amount of outdoor air required for each space is no longer based on the number of people in the space alone, as in the old standard, but a combination of the number of people and the floor are of the space. The new standard lists the amount of outdoor air required for the people and for the floor area for different types of spaces, as shown in Table 2.2. In the table R_p is the outdoor air flow rate per person [L/(s·person)] and R_a is the outdoor airflow rate per floor area [$L/(s \cdot m^2)$]. Using this standard the total outdoor ventilation rate supplied to the building is 11,100 L/s (23,500 cfm). This new standard will be used in a sensitivity study to determine the effect it has on the simulations.

Table 2.2. Parameters required for the new ventilation standard including: number of people, floor area, occupant factor and area factor.

Space	People	Area (m^2)	R_p	R_a
Lobby	20	702	2.5	0.3
Washroom	14	174	-	-
Office	1523	21757.5	2.5	0.3
Elevator	-	13.4	-	0.3
Stairs	-	69	-	0.3

2.1.4 Occupancy in the Office Building

The number of occupants chosen for the office building is also based on values from ASHRAE Standard 62 (2001). A value of 7 people/100 m^2 (7 people/1000 ft^2) is suggested for the office space, which results in a total of 1523 people. For the lobby a value of 30 people/100m^2 (30people/1000ft^2) is suggested. Based on the size of the given lobby this value gives an unreasonable number of occupants, so the occupancy level is set at 20 people, approximately one tenth the suggested number. Since the ventilation rate for the washroom is based on the number of people using the washrooms, an occupancy level has to be set for this space as well. It is assumed that one person will be using the washroom on each floor for a total of 14 people in the washrooms at all times during the day.

In order to simulate the heat gain from the people, the activity level of the occupants has to be set for each of the spaces. These values are chosen based on a table given in the computer program and are displayed in Table 2.3, along with the associated heat gains (sensible and latent) from each type of activity.

Table 2.3. Activity levels of the occupants in each zone.

Zone	Number of People	Activity	Associated Heat Gain
Office	1523	Seated, light work, typing	q_s = 75 W, q_l = 75W
Lobby	20	Seated, light work, typing	q_s = 75 W, q_l = 75W
Washroom	14	Seated at rest	q_s = 60 W, q_l = 40W

The occupancy schedule for each of the spaces can be found in Figures 2.5 and 2.6. These schedules show the fraction of the total number of people that are present in the space at each hour throughout the day. The building is unoccupied on weekends.

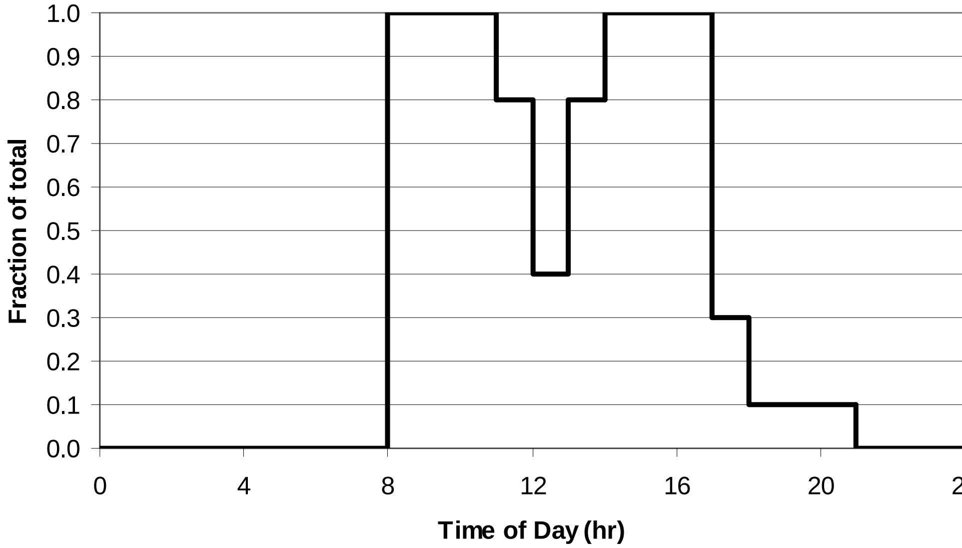


Figure 2.5. Occupancy schedule for the office and lobby spaces in the office building.

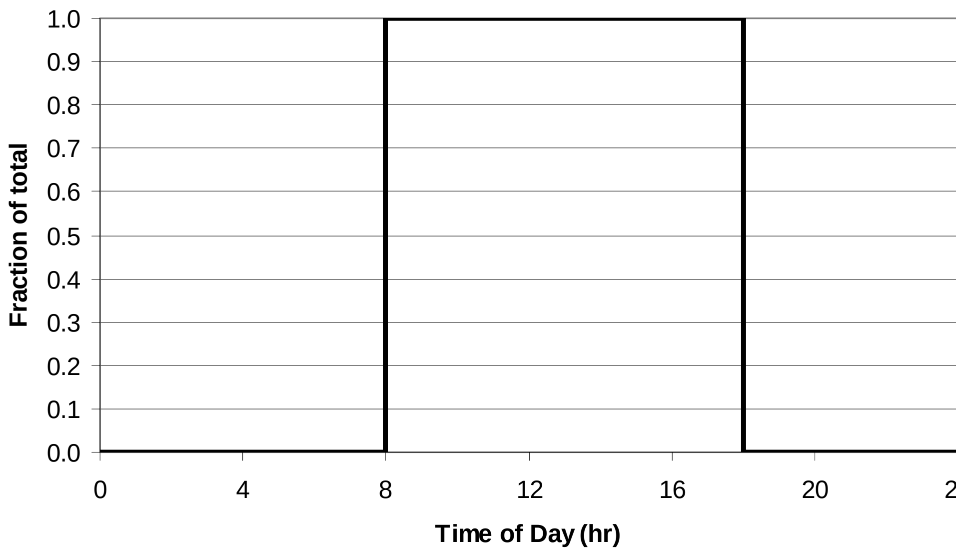


Figure 2.6. Occupancy schedule for the washrooms in the office building.

In order to calculate the thermal comfort levels of the people in the building, the clothing factor and metabolic rate of the people in each space has to be specified. For the office space the clothing factor is set at 1.0 clo (typical business suit) where 1 clo = 0.155 $m^2 \cdot C/W$ (0.880 $F \cdot ft^2 \cdot hr/BTU$) and the metabolic rate at 1.2 met (seated, light work) where 1 met = 58.2 W/m^2 (18.4 Btu/(hr·ft^2)). The clothing factor for the people in the lobby is also set at 1.0 clo and the metabolic rate at 1.7 met (people walking about). Another factor in the thermal comfort level is the relative velocity of the air which is set at 0.1 m/s. The clothing factor values, metabolic rate for the occupants in the office and lobby spaces and the relative velocity of the air are taken from ASHRAE (2005).

2.1.5 Equipment and Lighting in the Office Building

The values used by Dhital (1994) for the equipment and lighting are 27 W/m^2 (8.6 BTU/(hr·ft^2)) and 21.5 W/m^2 (6.8 BTU/(hr·ft^2)), respectively. The TRNSYS program gives a value of 230 W (785 BTU/hr) for a computer with color monitor, which would be the main type of equipment in an office building and is similar to the heat gain values given in ASHRAE (2005). To make this consistent with Dhital's numbers, 2554 computers would have to be used in the space. This seems very high for the number of people in the office, so instead it is assumed that there is one computer for every person for a total of 1537 computers. The receptionists in the lobby will also require computers so six 230W computers with color monitors are chosen. The schedule for the use of the computers is seen in Figure 2.7. The equipment is off on the weekends.

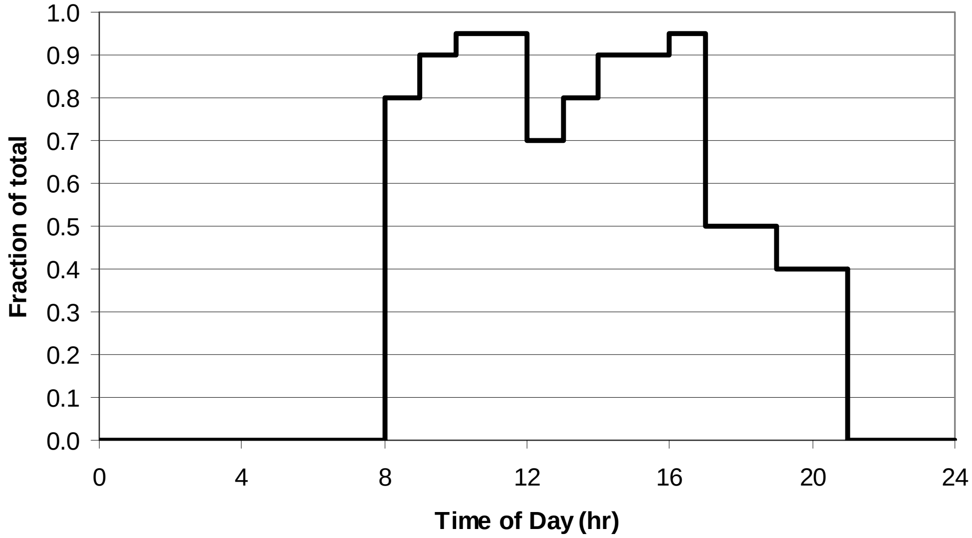


Figure 2.7. Computer schedule for the office and lobby spaces in the office building.

The heat gain from the lighting is selected from a table in the computer program and is chosen to be 19 W/m^2 (6.0 BTU/(hr·ft^2)), which is similar to the value of 21.5 W/m^2 used by Dhital (1994). The heat gain from the lights is assumed to be 40% convective. The schedule for the lighting in each of the conditioned spaces can be seen in Figures 2.8 and 2.9. The lights are left on at 10% during the evenings and weekends.

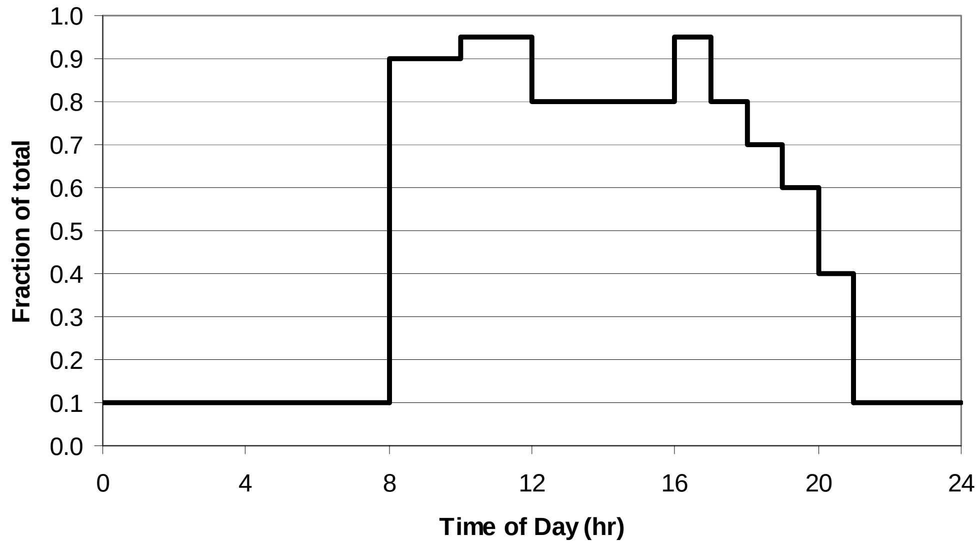


Figure 2.8. Lighting schedule for the office space in the office building.

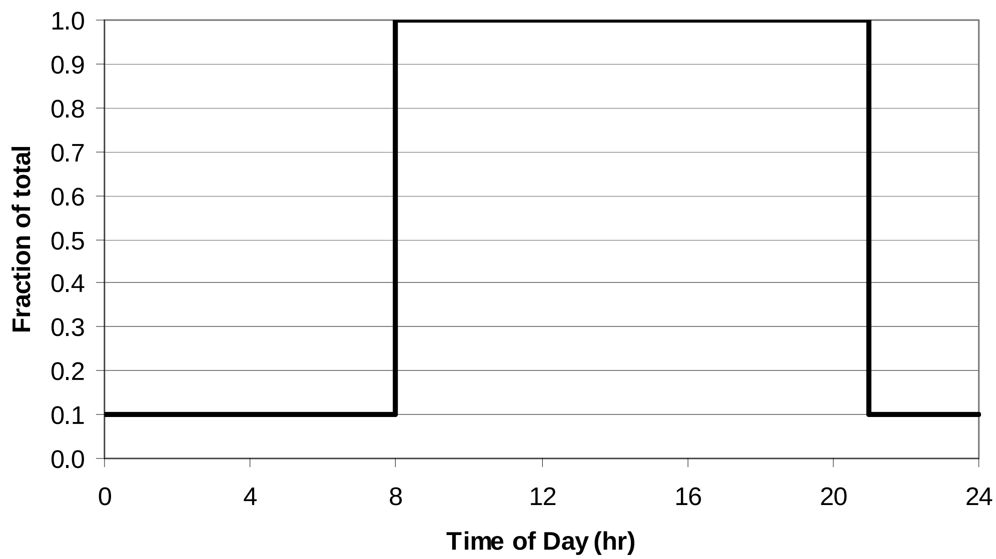


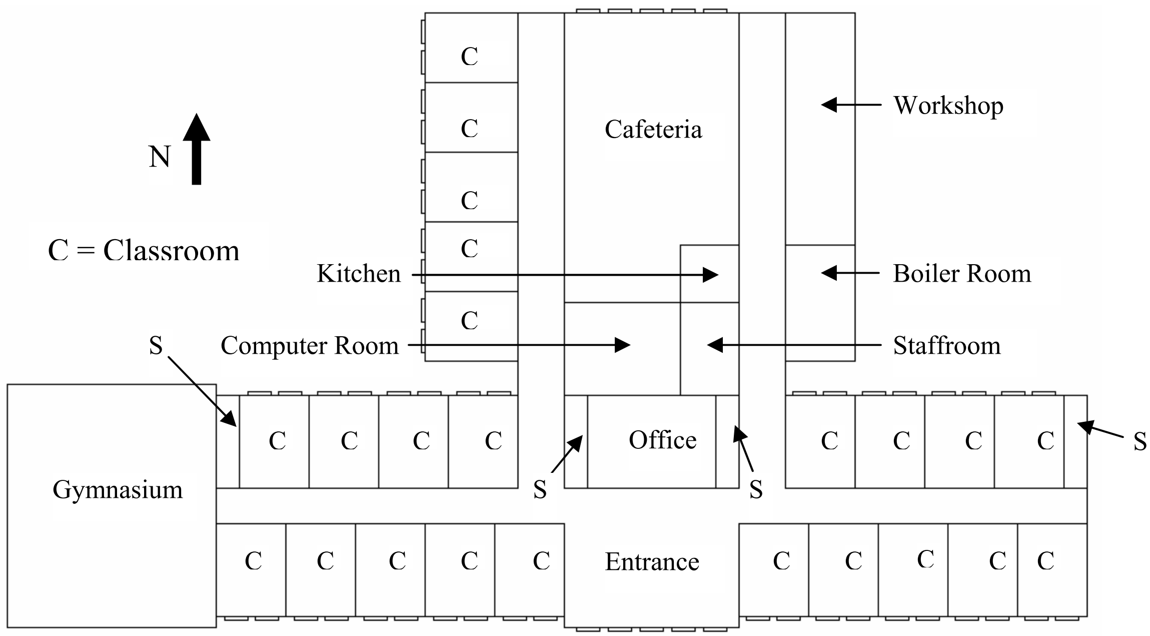
Figure 2.9. Lighting schedule for the lobby, washrooms, elevator and stairwells in the office building.

2.2 THE SCHOOL BUILDING

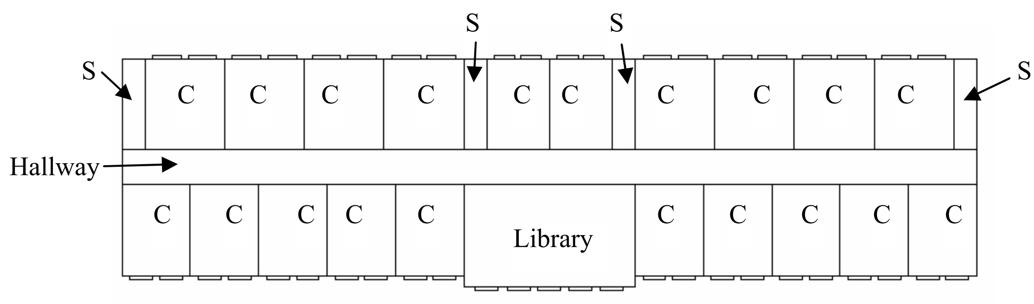
The floor plan chosen for the school is based on the floor plan of the LaGrange Middle School located in LaGrangeville, NY. This floor plan was obtained from their website (LaGrange Middle School, 2000). Many of the details of the school are taken out to make it a more simple design, which would enable it to be programmed into TRNSYS.

2.2.1 Floor Plan of the School

The school building, as seen in Figure 2.10 has a total floor area of 4437m^2 (47760ft^2). The first floor consists of 23 classrooms (C), 4 stairwells (S), a cafeteria, workshop, boiler room, staffroom, kitchen, computer room, the main office and several hallways. The second floor covers only the front half of the building and consists of 20 classrooms, 4 stairwells and the library. Each floor is 2.7m (9ft) high with a 1m (3.3ft) plenum above it. The gymnasium is located on the west end of the school and is 7.5m (25ft) high. The floor plan from the website did not have dimensions on it, so the size of each room is chosen based on the design occupancy and the occupancy density given in ASHRAE Standard 62 (2001). The dimensions for each room are given in Table 2.4. As with the office building, each room in the school is in a thermal zone with all of the classrooms being in one thermal zone.



(a)



(b)

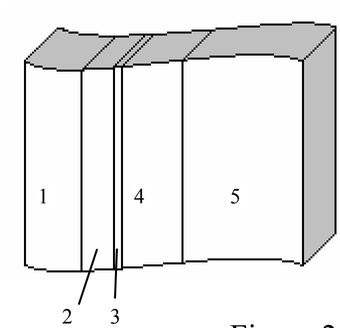
Figure 2.10. Floor plan of the first level (a) and the second level (b) of the school.

Table 2.4. Dimensions of each space in the school building.

Space	Dimensions
Gymnasium	18m x 21m
Cafeteria	15m x 25m
Kitchen	5m x 5m
Computer Room	10m x 8m
Staffroom	5m x 8m
Office	11m x 8m
Entrance	15m x 9m
Workshop	6m x 20m
Boiler Room	6m x 10m
Library	15m x 9m
Stairwells	2m x 8m
First Floor Classrooms	6m x 8m
Second Floor Classrooms:	
South Facing	6m x 8m
North Facing (Large)	7m x 8m
North Facing (Small)	5.5m x 8m

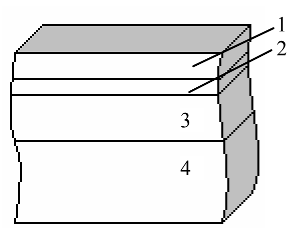
2.2.2 Construction of the School

The exterior walls of the school are made of 10 cm (3.94 in) of brick on the outside, a 5 cm (1.97 in) air space, 2 cm (0.79 in) of plywood, 10 cm (3.94 in) of expanded polystyrene insulation and 20 cm (7.87 in) concrete blocks on the interior, as shown in Figure 2.11. The U value for this wall is U = 0.166 W/(m^2·K) (R = 34.1 h·ft^2·F/BTU). The roof is of the same construction as the roof of the office building and has a U-value of 0.496 W/(m^2·K) (R = 11.5 h·ft^2·F/BTU), as shown in Figure 2.12. The windows are triple pane clear glass and are located in the classrooms, cafeteria, library and front entrance. They have a U value of 1.3 W/(m^2·K) (R = 4.4 h·ft^2·F/BTU). The interior walls are made of 20 cm (7.87 in) thick concrete blocks. The ceiling between the space and the plenum above is made of 2 cm (0.79 in) acoustic tile. The main floor is made of 20 cm (7.87 in) of concrete and 5 cm (1.97 in) of polystyrene insulation. The floor of the second storey is made of 10 cm (3.94 in) of concrete. The thermal capacitance is set at 12 times the space volume for all of the spaces that have furniture in them and 1.2 times the space volume for the spaces that do not have any furniture.



Material	Thickness (cm)
1. Brick face exterior	10
2. Air space	5
3. Plywood	2
4. Expanded polystyrene insulation	10
5. Concrete block interior	20

Figure 2.11. Cross-section of the exterior walls of the school ($U = 0.166$ W/(m^2·K), $R = 34.1$ h·ft^2·F/BTU).



Material	Thickness (cm)
1. Gravel (top)	2.5
2. Built-up roofing	1
3. Expanded polystyrene insulation	5
4. Concrete (bottom)	10

Figure 2.12. Cross-section of the roof of the school ($U = 0.496$ W/(m^2·K), $R = 11.5$ h·ft^2·F/BTU).

2.2.3 Infiltration and Ventilation in the School

The infiltration rate through the exterior walls is the same as for the office building, 800 cm^3/(s·m^2) (0.157 cfm/ft^2) but the number of ach varies depending on the surface area of the exterior wall and the volume of each space. The classrooms, which account for the majority of the school building, have an infiltration of 0.4 ach. The ventilation rates for the school are from ASHRAE Standard 62 (2001). These values can be found in Table 2.5. The total outdoor ventilation rate supplied to the school is constant at 20,426.5 L/s (43,281.3 cfm). Similar to the office building, the ventilation system is turned on two hours prior to occupants entering the building (i.e., 7:00) and is shut off two hours after the last occupants leave (i.e., 19:00).

Table 2.5. Ventilation rates for each space in the school.

Space	Ventilation Rate
Gymnasium	10 L/(s·person)
Cafeteria	10 L/(s·person)
Kitchen	8 L/(s·person)
Computer Room	10 L/(s·person)
Staffroom	8 L/(s·person)
Office	10 L/(s·person)
Workshop	10 L/(s·person)
Library	8 L/(s·person)
Classroom	8 L/(s·person)
Hallway	0.50 L/(s·m^2)
Stairwells	0.50 L/(s·m^2)

2.2.4 Occupancy in the School

The occupancy levels for the school are chosen based on a typical high school which has 30 students and one teacher per classroom. There are 43 classrooms as well as three classes in the gym, one class in the library, one class in the computer lab and two classes in the workshop at all times throughout the school day (9:00-12:00 and 13:00-16:00). This gives a total of 1500 students. On top of this, there are 51 teachers, 5 office personnel and 2 librarians in the school. The occupancy level for each space during the school day is given in Table 2.6. The school is unoccupied on the weekends.

Table 2.6. Occupancy levels in the school during school time.

Space	Occupancy	Explanation
Classroom x 43	31	30 students + 1 teacher
Gymnasium	93	3 classes + 3 teachers
Library	33	1 class, 1 teacher + 2 librarians
Computer Lab	31	1 class + 1 teacher
Workshop	63	2 classes + 3 teachers
Main Office	5	office personnel

During lunchtime (12:00-13:00) half of the students leave the building while one quarter of the students (375) go to the cafeteria and the remaining 375 students go to the gymnasium. All of the staff members go to the staffroom and an additional 5 people come in to the kitchen to cook. After school (16:00-17:00) there are 30 students and 3 teachers in the gym and 2 people in the office. The activity level for the occupants within each space is chosen from a menu within the computer program, and is shown in Table 2.7.

Table 2.7. Activity levels for the occupants in each space of the school, during maximum occupancy.

Space	Max # of People	Activity	Associated Heat Gain
Gymnasium	375	Heavy work, athletics	q_s = 185W, q_l = 340W
Cafeteria	375	Seated, eating	q_s = 75W, q_l = 95W
Kitchen	5	Light benchwork	q_s = 100W, q_l = 130W
Computer Room	31	Seated, light work, typing	q_s = 75W, q_l = 75W
Staffroom	58	Seated, eating	q_s = 75W, q_l = 95W
Office	5	Seated, light work, typing	q_s = 75W, q_l = 75W
Workshop	63	Light machine work	q_s = 100W, q_l = 205W
Library	33	Seated, light work, typing	q_s = 75W, q_l = 75W
Classroom	1333	Seated, very light writing	q_s = 65W, q_l = 55W

The clothing and metabolic rates for the people in each of the spaces can be seen in Table 2.8. The relative velocity of the air is set at 0.1 m/s. These values are again taken from ASHRAE (2005).

Table 2.8. Clothing and metabolic rates for occupants in the school building.

Space	Clothing Factor (clo)	Explanation	Met Rate (met)	Explanation
Office	0.5	Light weight trousers, short sleeve shirt	1.2	Seated, light work
Computer Room	0.5	Light weight trousers, short sleeve shirt	1.2	Seated, light work
Staffroom	0.5	Light weight trousers, Short sleeve shirt	1	Seated, relaxed
Kitchen	0.6	Work shirt and trousers	2	Cooking
Cafeteria	0.5	Light weight trousers, short sleeve shirt	1	Seated, relaxed
Workshop	0.6	Work shirt and trousers	1.8	Machine work
Classroom	0.5	Light weight trousers, short sleeve shirt	1.2	Seated, light work
Library	0.5	Light weight trousers, short sleeve shirt	1	Seated, relaxed
Gymnasium	0.2	Shorts and t-shirt	4	Calisthenics/exercise

2.2.5 Equipment and Lighting in the School

The school building contains several forms of equipment, mainly computers and kitchen appliances. The computers are all PCs with color monitor and have a heat gain of 230W (785BTU/hr). There are two computers in the main office, two in the library and 30 in the computer room. The computers in the computer room and the library are on from 9:00-16:00 and in the office from 9:00-17:00.

The remaining equipment is listed in Table 2.9 along with the time that it is in use and the associated radiant, convective and if appropriate, latent heat gains. The heat gain values are chosen from ASHRAE Fundamentals (2005) and then entered into the computer simulation. ASHRAE (2005) gives the overall heat gain but the computer program requires the radiant and convective portions so the percentage of each has to be determined. Table 2.10 shows the radiant portions of the heat gain of some of the equipment, as given by ASHRAE (2005). All remaining equipment is assumed to be 70% radiant.

Table 2.9. Heat gains from the equipment in the school building.

Office	**Time**	**Radiant (W)**	**Convective (W)**	**Latent (W)**
Photocopier	all	75	225	0
Staffroom	**Time**	**Radiant (W)**	**Convective (W)**	**Latent (W)**
Microwave (residential)	12-1p	980	420	0
Coffeemaker (10cups)	12-1p	735	315	450
Kitchen	**Time**	**Radiant (W)**	**Convective (W)**	**Latent (W)**
Deep Freeze (large)	all	378	162	0
Large Microwave	12-1p	1841	789	0
Convection Oven (gas)	12-1p	334	1336	0
Fryer (x2) (deep fat)	12-1p	481.6	638.4	0
Griddle (large)	12-1p	873	1067	1080
Coffeemaker	12-1p	735	315	450
Fridge (0.5m^3)	all	241.5	103.5	0
Cafeteria	**Time**	**Radiant (W)**	**Convective (W)**	**Latent (W)**
Pop Machine (x2)	all	1344	576	0
Vending Machine (x2)	all	385	165	0
Microwave (residential)	12-1p	980	420	0
Library	**Time**	**Radiant (W)**	**Convective (W)**	**Latent (W)**
Photocopier	all	75	225	0

Table 2.10. Radiative portion of the equipment heat gains.

Equipment	**Radiant Portion**
Griddle	0.45
Fryer	0.43
Oven	0.20
Copier	0.25

As with the office building, the heat gain from the lighting is chosen to be 19 W/m^2 with 40% being convective. The lights in each space are on when that space is occupied, and shut off when the space is not occupied. All lights are off on the weekends.

2.3 HVAC SYSTEM

In this thesis, two HVAC systems will be studied - one without an energy wheel (termed the base case) and one with an energy wheel. These systems will be used in both the office and school buildings. A schematic of the HVAC system in the base case is shown in Figure 2.13.

In the base case (Figure 2.13), air is taken from outside and passed through an economizer (E), which controls the fraction of outdoor air (OA) and recirculation air (RA) delivered to the building. Under normal operating conditions the fraction of OA is set at 20% of the total supply air (SA). If the outdoor temperature and humidity are favorable, the economizer will increase the percentage of OA being used. This will reduce the amount of energy required for cooling during moderate weather conditions. As shown in Figure 2.13, the mixture of OA and RA is conditioned by passing through the cooling (C) and heating (H) units, as needed. The conditioned air is then supplied to the space by the supply fan (SF). The exhaust fan (EF) extracts air from the building and a portion of this air is used for recirculation (RA) and the rest is exhausted from the building (EA). Both the SF and the EF are constant volume fans.

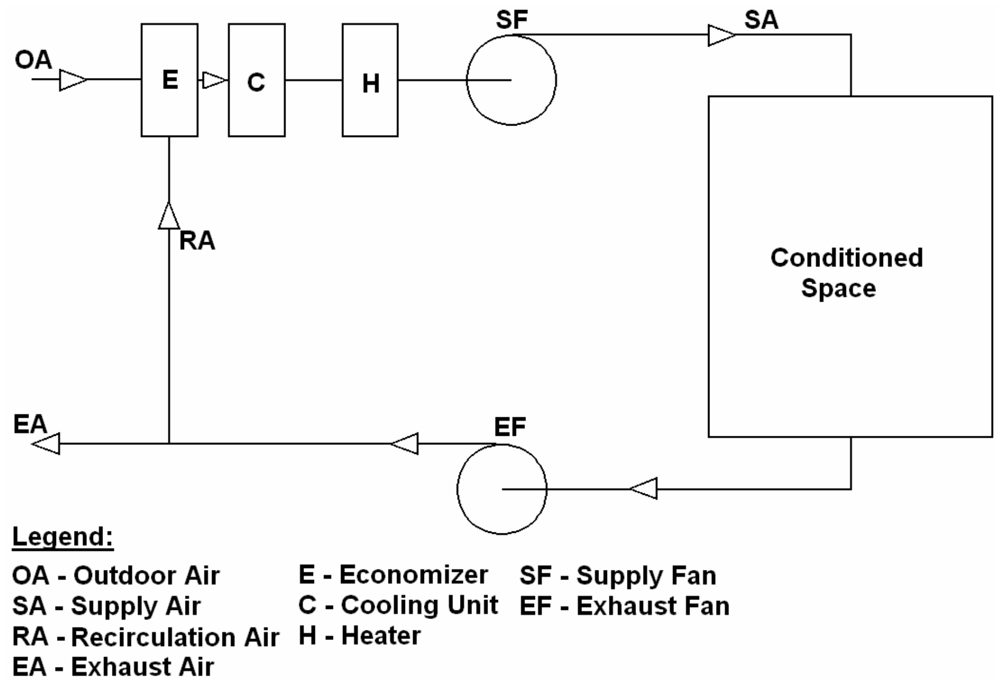


Figure 2.13. Schematic of the HVAC system for the base case simulations.

The HVAC system for the simulation with an energy wheel is similar to the base case. The only differences are the addition of an energy wheel and a bypass damper to control the wheel during part load conditions. The modifications to the base HVAC system are highlighted in light grey in Figure 2.14. The energy wheel transfers heat and moisture between the OA entering the system and the EA leaving the system. This air exiting from the supply side of the energy wheel is then mixed with RA, as in the base case.

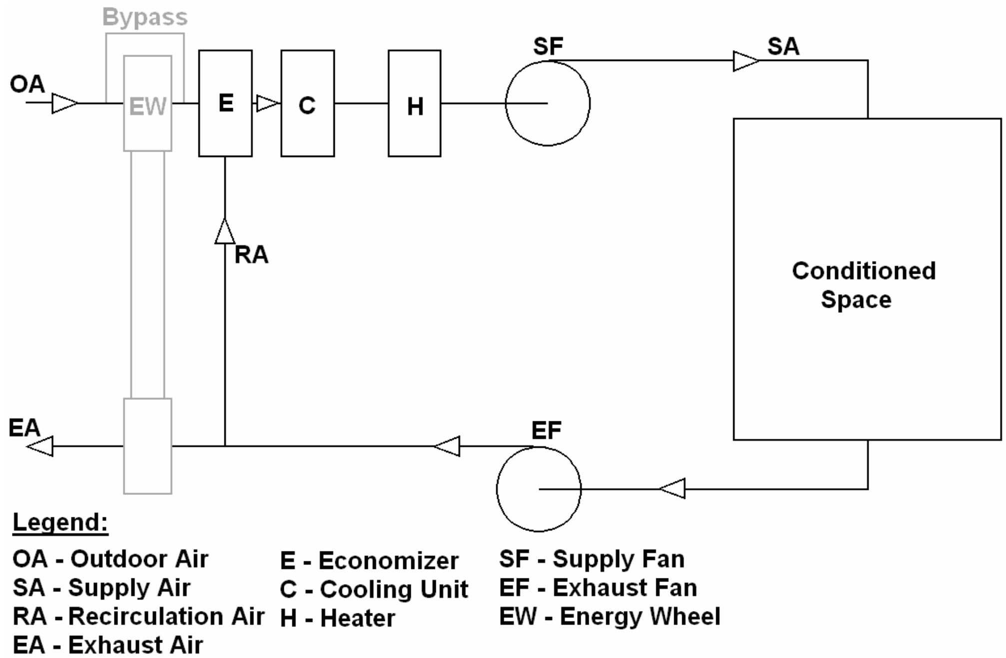


Figure 2.14. Schematic of the modified HVAC system, including an energy wheel.

2.3.1 Economizer

As mentioned previously, the normal percentage of OA is 20% of the SA. There are some circumstances, however, where it is beneficial to use a larger percentage of OA. This is when the economizer is utilized. The economizer uses dampers, which open and close to increase, or decrease, the fraction of OA supplied to the conditioned space. These dampers are controlled by the outdoor temperature. The schedule for the economizer is shown in Figure 2.15, with X being the fraction of RA in the SA (normally X = 80%). When the outdoor temperature reaches a certain value, T_{bypass}, the economizer begins to decrease the amount of RA, thus increasing the percentage of OA. The fraction of RA reaches zero when the outdoor temperature is equal to the design supply temperature, $T_{s,design}$. When the OA temperature is equal to, or greater than the indoor air temperature, T_{in}, the amount of RA is set to 80% of the SA. The amount of recirculation air is never more than 80% when the building is occupied because at least 20% OA is needed to meet the ventilation requirements of the space.

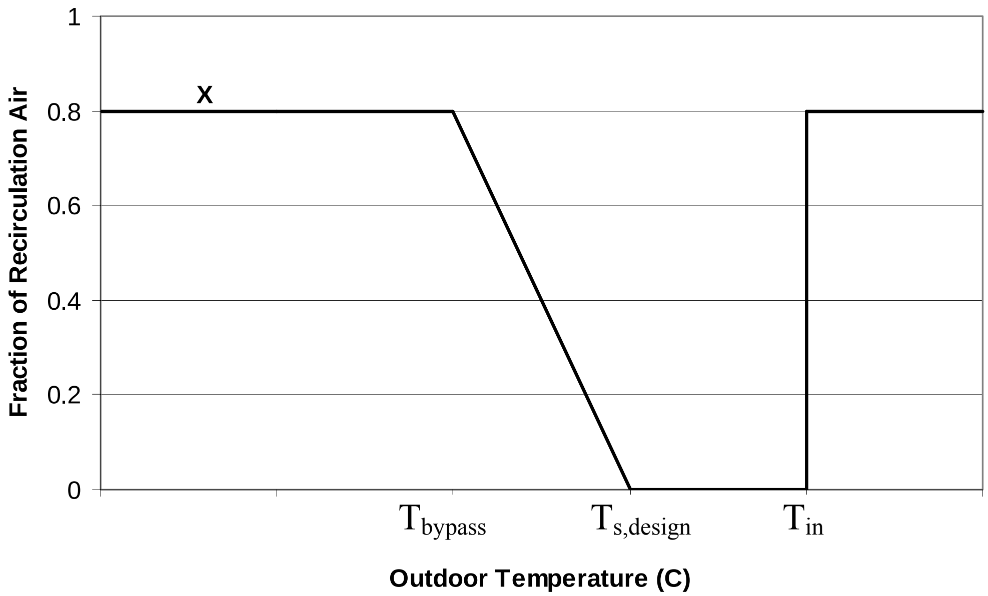


Figure 2.15. Schedule for the economizer as a function of the outdoor temperature.

The temperature at which the economizer begins to increase the percentage of outdoor air, T_{bypass} is calculated as (Simonson et al., 2000),

$$T_{bypass} = \frac{T_{s,design} - X_{max} T_{in}}{1 - X_{max}}, \tag{2.1}$$

where X_{max} is equal to 0.8. The design supply temperature, $T_{s,design}$, is the outdoor air temperature at which the cooling load of the space will be just satisfied with no additional cooling. For this study, $T_{s,design}$ = 14°C (57°F) for both buildings, which was determined by a quick hand calculation, and proved acceptable with the simulations. It was found that a SA temperature of 14°C will sufficiently remove any heat generated in the space by occupants and equipment. When the outdoor temperature is in the range between T_{bypass} and $T_{s,design}$, the cooling load can be met with an appropriate fraction of

RA (X) and OA (1-X). During these conditions the fraction of RA (X) can be calculated by,

$$X = \frac{T_{s,design} - T_{outdoor}}{T_{in} - T_{outdoor}}. \tag{2.2}$$

After passing through the economizer (E) the mixed air enters the cooling and heating units. These units are controlled by a thermostat which is connected to the space being conditioned. The thermostat is an on/off controller with heating setback. The set temperatures are 22°C (72°F) for heating and 24°C (75°F) for cooling. A nighttime heating setback is utilized and set at 18°C (64°F). This is accomplished using a forcing function that outputs a value of 1 during the night and 0 during the day. This is linked to the night setback control input of the thermostat and is multiplied by the setback temperature difference. During the night the setback difference is 4°C (8°F) and during the day it is 0°C (0°F).

2.3.2 Cooling Unit

When the temperature in the building is above 24°C the SA needs to be cooled. This is done using a cooling coil, similar to the one shown in Figure 2.16, from the Air-Conditioning-and-Refrigeration-Guide website (2005). A cooling unit consists of four major parts: the compressor, condenser, evaporator and the expansion valve. These components are connected with piping which contains a refrigerant. In Figure 2.16, the refrigerant is shown in dark grey when it is warm, light grey when it is cool, as a solid line when it is a liquid and a broken line when it is a vapor. The refrigerant enters the compressor as a vapor at low pressure and temperature. Here the vapor is compressed and exits at a higher pressure, and at a temperature about 30°C (54°F) above the outdoor air temperature. The refrigerant then travels through the condenser coil. Outdoor air is passed across the coil and heat is transferred from the refrigerant to the air. When the heat is removed from the refrigerant it begins to condense. The refrigerant exits the condenser as a high pressure liquid.

After leaving the condenser the refrigerant enters an expansion valve, where it experiences a large pressure drop, from approximately 1,550 kPa (225 psi) to 480 kPa (70 psi). As this is a constant enthalpy process, there is a large temperature drop, from about 43°C (110°F) to 4°C (40°F). The cooled refrigerant can be used to cool another fluid as it evaporates in the evaporator coil. The fluid that is cooled can be air (as shown in Figure 2.16 for a direct expansion cooling unit) or another fluid as in the case of a chiller. In this thesis, a direct expansion cooling unit is selected because such a component is simpler to implement in TRNSYS. As the focus of this thesis is on the impact of an energy wheel on energy consumption and PAQ, the selection of a direct expansion cooling unit or chiller is not considered critical. The actual energy consumption or PAQ may change slightly with different cooling units, but the relative impact of the energy wheel is expected to be similar. Therefore, the cooling unit, in Figure 2.13 can be considered a direct expansion coil or a cooling coil with chilled water from a remote chiller.

As the mixture of OA and RA passes through the coiling coil (C) it is cooled. As well, moisture from the air will condense on the surface of the cooling coil, if the air is cooled below the dew point temperature. The now cooled, and possibly dehumidified air can be supplied to the building.

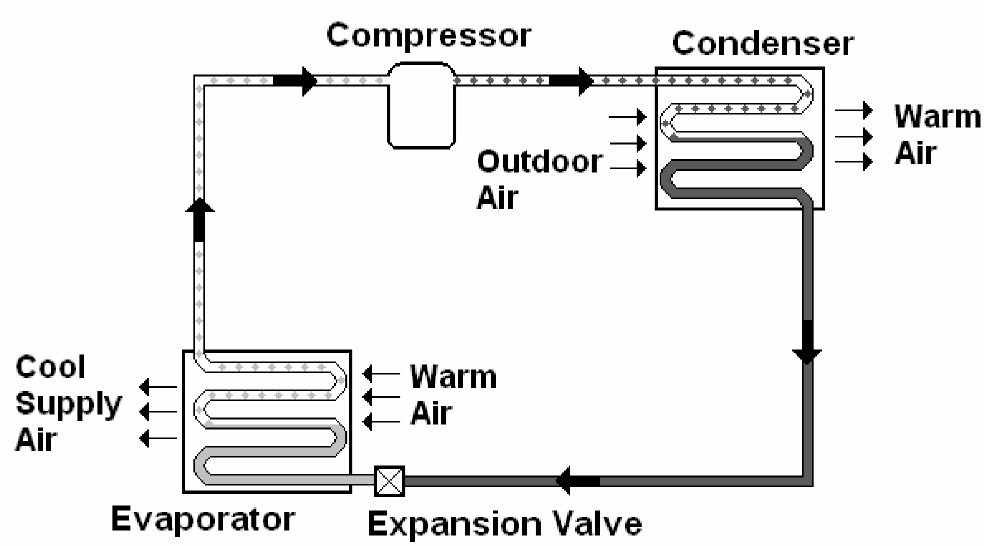


Figure 2.16. Schematic of a direct expansion cooling unit (Picture from the Air-Conditioning-and-Refrigeration-Guide website, 2005).

The TRNSYS program allows the cooling load to be calculated internally in the building model by specifying a set point temperature and a maximum cooling power, or externally from the building by modeling all of the components of the HVAC system, including the cooling unit described above. In this thesis the simulations are first performed for each building in each city using the internal cooling load calculation to determine the maximum cooling load required for the building. The simulations are then run using the external HVAC components and the information gathered from the first simulations is used to size the cooling coil in the system.

The cooling coil used in the simulations requires two information files: one file containing the total capacity and power as a function of the outdoor dry bulb temperature, the indoor wet bulb temperature and the evaporator flow rate and a second file containing the ratio of sensible to total cooling capacity as a function of the indoor dry bulb and wet bulb temperatures and the evaporator flow rate. It is from these files that the cooling unit determines how much cooling and dehumidifying occurs. The program came with sample files which have to be scaled to fit the requirements of this simulation because the cooling unit used in the simulation requires a larger capacity then the sample files give. These files are scaled based on the information obtained from the internal cooling load calculations. The cooling coil is given a coefficient of performance (COP) of 2.7 based on ASHRAE Standard 90.1 (2004). The COP is the ratio of the energy removed from the air by the cooling coil to the power consumed by the cooling coil.

2.3.3 Heating Unit

When the air temperature in the space is below 22°C (72°F) during the day or 18°C (65°F) during the night the SA air must be heated. This is accomplished using a boiler. Inside the boiler, natural gas is burned in a combustion chamber. The process of combustion produces heat, which is used to heat water. The hot water then passes through a heat exchanger where it transfers heat to the SA. Some of the heat produced by combustion is lost to the surroundings as the byproducts of the combustion process are vented through a flue to the outdoors. It is assumed that the efficiency of the boiler is 79%. The heated air is now ready to be supplied to the space.

As with the cooling coil, the boiler is sized using the internal heating load calculations performed by the building model in TRNSYS. To calculate the heating load the set point temperature and the maximum heating power are required by the program. The maximum heating load is then used in the simulations as the maximum heating rate of the boiler component in the HVAC system. The ventilation system is run two hours before the start of occupation until two hours after the end of occupation but sometimes heating and cooling is required outside of these hours. When this occurs the supply air is provided to the building using 100% RA (0% OA).

2.3.4 Supply and Exhaust Fan

The fans are another major component in the HVAC system. In this system there are two constant volume fans, a supply fan, located just before the space and an exhaust fan located just after the space. The main purpose of a fan is to create airflow in the ducts. The flow rate created by the fans is 79,000 L/s (167,392 cfm) for the office building and 102,000 L/s (216,126 cfm) for the school. The supply fan has a pressure drop of 1250 Pa (5 inH_2O) and the exhaust fan has a pressure drop of 500 Pa (2 inH_2O). Both fans have an efficiency of 70%. These values are based on the simulations run by Dhital (1994). The fan power is determined from the flow rate, pressure drop and efficiency of the fan. Together with the heating and cooling units and the economizer the fans make up the HVAC system for the base case simulations.

2.3.5 Energy Wheel

The sensible and latent effectivenesses of the energy wheel and the pressure drop across the wheel are the most important parameters for modeling an energy wheel in an HVAC system. The outdoor air correction factor and exhaust air transfer rate are also important but are assumed to be equal to 1 and 0%, respectively in this thesis. Therefore, the outdoor ventilation rate is the same in the base case as in the energy wheel case. In this thesis the pressure drop across the energy wheel is assumed to be 200 Pa (0.8 inH_2O) on both the supply and exhaust sides. The effectivenesses of the energy wheel are assumed to be constant throughout the year or changing as the outdoor temperature and humidity change throughout the year.

The constant effectiveness is chosen to be 70% for both sensible and latent heat transfer. An effectiveness of 70% is representative of a commercial energy wheel. The variable effectivenesses are also chosen to be 70% when the wheel is operating under ARI summer conditions (ANSI/ARI Standard 1060, 2001), as listed in Table 2.11. If the winter conditions had been chosen the effectivenesses would be set to 72.7%, which would not make a large impact on the results.

Table 2.11. ARI summer and winter test conditions.

	Heating	Cooling
Entering Supply Airflow Temperature		
(a) Dry-Bulb	1.7°C (35°F)	35°C (95°F)
(b) Wet-Bulb	0.6°C (33°F)	26°C (78°F)
Entering Exhaust Airflow Temperature		
(a) Dry-Bulb	21°C (70°F)	24°C (75°F)
(b) Wet-Bulb	14°C (58°F)	17°C (63°F)

During different temperature and humidity operating conditions, the effectiveness of the energy wheel will change as the operating condition factor (H*) changes according to the following correlations developed by Simonson and Besant (1999b):

$$\varepsilon_s = \frac{NTU_o}{1+NTU_o}\left(1-\frac{1}{7.5Cr_o^*}\right)-\left[\frac{0.26\left(\frac{Cr_o^*}{Wm^2Crm_o^*}\right)^{0.28}}{7.2\left(Cr_o^*\right)^{1.53}+\frac{210}{\left(NTU_o\right)^{2.9}}-5.2}+\frac{0.31\eta}{\left(NTU_o\right)^{0.68}}\right]H^* \quad (2.3)$$

and,

$$\varepsilon_l = \frac{NTU_o}{1+NTU_o}\left(1-\frac{1}{0.54\left(Cr_{mt,o}^*\right)^{0.86}}\right)\left(1-\frac{1}{\left(NTU_o\right)^{0.51}\left(Cr_{mt,o}^*\right)^{0.54}H^*}\right). \quad (2.4)$$

In equations (2.3) and (2.4), the overall number of transfer units (NTU_o), overall matrix heat capacity ratio (Cr_o^*) and the overall matrix moisture capacity ratio (Crm_o^*) are parameters of the energy wheel and will be constant (independent of the operating temperature and humidity) for a given energy wheel. The maximum moisture content of the desiccant within the energy wheel (Wm) and the fraction of phase change energy that is delivered directly to the air (η) can also be assumed to be independent of the operating temperature and humidity. H* depends on outdoor temperature and humidity and is calculated by

$$H^* = \frac{\Delta H_l}{\Delta H_s} = 2500\frac{\Delta W}{\Delta T} = 2500\frac{\left(W_{s,i}-W_{e,i}\right)}{\left(T_{s,i}-T_{e,i}\right)}, \quad (2.5)$$

where W is humidity ratio, T is temperature, subscript s,i refers to the inlet on the supply side of the wheel and subscript e,i refers to the exhaust side inlet. H* may vary over a range of -60 to +60 in HVAC systems but has a practical range of importance of -6 to +6 for air-to-air energy recovery (Simonson and Besant 1999a).

The storage of moisture in the desiccant is also affected moderately by the operating temperature and humidity. It is calculated by

$$Cr^*_{mt,o} = \left(Crm^*_o\right)^{0.58}\left(Wm\right)^{0.33}\left(\left.\frac{\partial u}{\partial \phi}\right|_{\phi_{ave}}\right)^{0.2}\left(Cr^*_o\right)^{1.13}\left(\frac{e^{\left(\frac{1482}{T_{ave}}\right)}}{47.9} - 1.26\left(\phi_{ave}\right)^{0.5}\right)^{4.66}, \quad (2.6)$$

where T_{ave} is $(T_{s,i}+T_{e,i})/2$ in K and ϕ_{ave} is $(\phi_{s,i}+\phi_{e,i})/2$. The slope of the sorption curve at the average operating humidity is

$$\left(\left.\frac{\partial u}{\partial \phi}\right|_{\phi_{ave}}\right) = \frac{\left(u_{s,i} - u_{e,i}\right)}{\left(\phi_{s,i} - \phi_{e,i}\right)}, \quad (2.7)$$

where u is the mass fraction of water in the desiccant. The energy wheel used in this thesis uses a silica gel desiccant which has a linear sorption curve and therefore the slope of the sorption curve will not change with humidity.

To determine the parameters of the energy wheel that will give sensible and latent effectivenesses of 70% at ARI summer conditions, equations (2.3) through (2.7) are solved iteratively to determine NTU_o, Cr^*_o and Crm^*_o, with the restrictions that Cr^*_o = NTU_o and $Crm^*_o = Cr^*_o/4$. The resulting parameters for the energy wheel are shown in Table 2.12 and the calculated effectivenesses over the entire year in Saskatoon, Vancouver, Phoenix and Tampa are presented in Figures 2.17 through 2.20. These figures are developed assuming that the indoor conditions are the ARI summer conditions from June 1st to September 30th and the ARI winter conditions for the rest of the year. The effect of the outdoor conditions on the sensible and latent effectivenesses can be seen as they vary throughout the year. The sensible and latent effectivenesses have been capped at 0% and 100%.

Table 2.12. Energy Wheel Parameters.

Parameter	Value
NTU_o	3.295
Cr_o*	3.295
Crm_o*	0.824
Wm	0.5
η	0.05

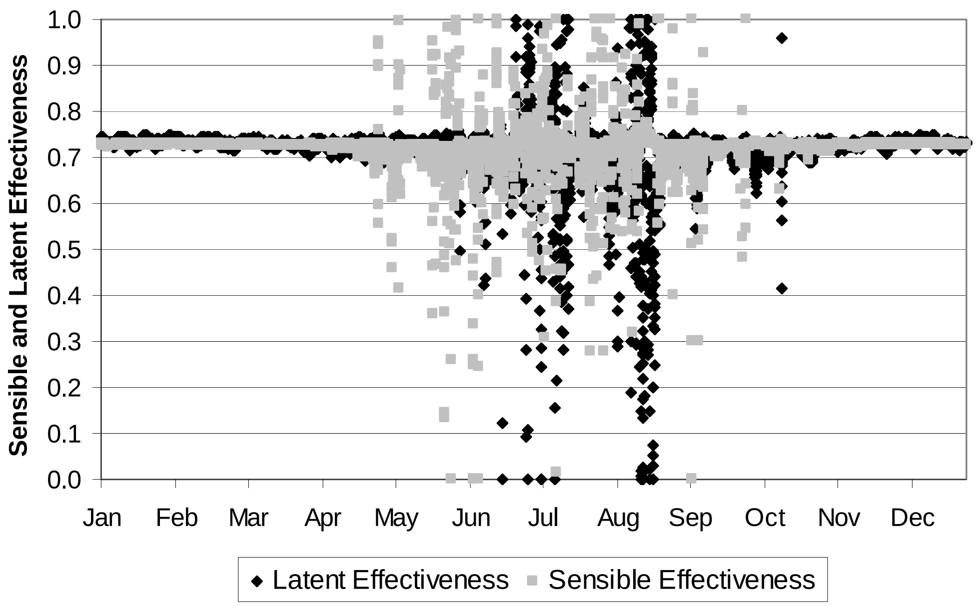


Figure 2.17. Sensible and latent effectiveness in Saskatoon throughout the year.

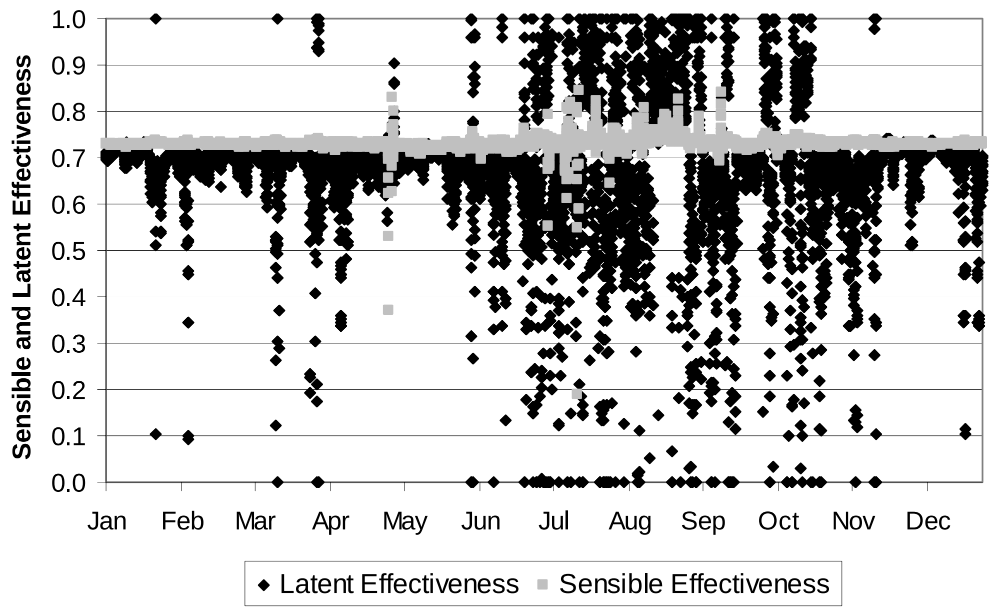


Figure 2.18. Sensible and latent effectiveness in Vancouver throughout the year.

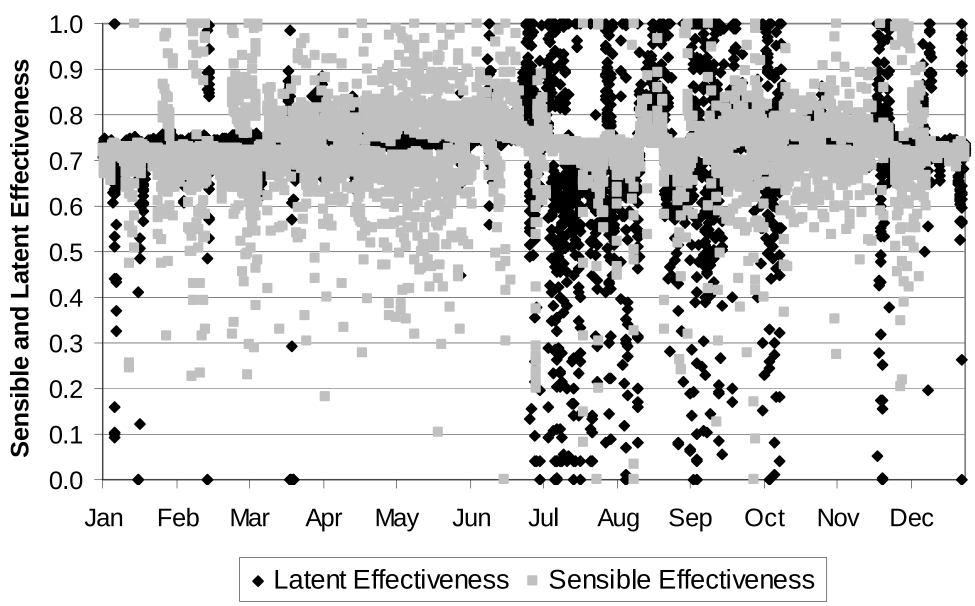


Figure 2.19. Sensible and latent effectiveness in Phoenix throughout the year.

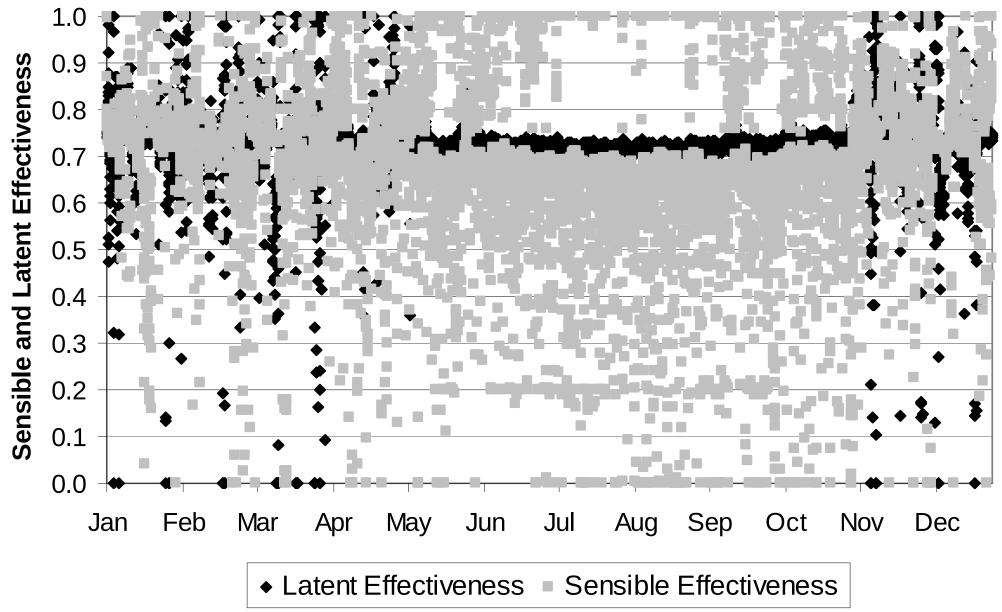


Figure 2.20. Sensible and latent effectiveness in Tampa throughout the year.

As mentioned earlier, energy wheels can help to significantly reduce the energy required to condition ventilation air for buildings. However, there are certain times in the year and certain climates where the use of an energy wheel can actually increase the building cooling demands. During these conditions, it is beneficial to reduce the amount of energy that the energy wheel transfers between the supply and exhaust air streams. These are called part load operating conditions. An example of these conditions is shown in Figure 2.21. Here the outdoor temperature is 18°C (65°F), which is between the design supply temperature and the indoor space temperature. The exhaust air coming from the space is at 22°C (72°F). Since the exhaust stream is hotter, heat will be transferred from the exhaust air to the outdoor supply air. When the outdoor air exits the energy wheel it will be at a temperature of 20.8°C (69.4°F) when the effectiveness is 70%, which is too hot to supply to the space. This air must then be cooled from a temperature of 20.8°C to the design supply temperature (14°C (57°F) in this thesis), whereas without the energy wheel the temperature would only have to be cooled from 18°C to the design supply temperature. The energy wheel has therefore increased the amount of cooling required in this situation.

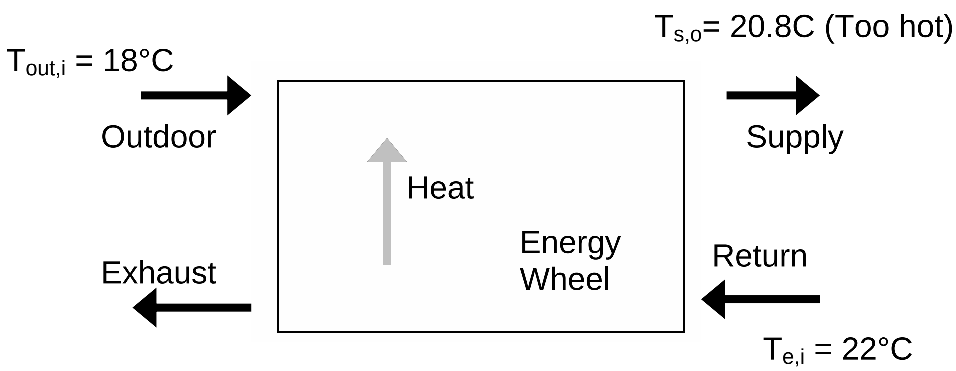


Figure 2.21. Schematic of the energy wheel during part load operating conditions.

In order to avoid overheating the supply air, the rate of heat transfer between the supply and exhaust air must be controlled. There are two standard practices for controlling the heat transfer rate of an energy wheel, wheel speed and bypass (Simonson et al. 2000a and 2000b). For this study, the energy wheel will be controlled by a bypass, with the addition of recirculated air (RA) control under some conditions. This is done with an economizer bypass by decreasing the effectiveness and then increasing the amount of outdoor air. This is accomplished using the schedule in Figure 2.22. The energy wheel will be controlled by decreasing the effectiveness until it is equal to 0%. At this point the percentage of outdoor air will be increased until it reaches 100% (0% RA).

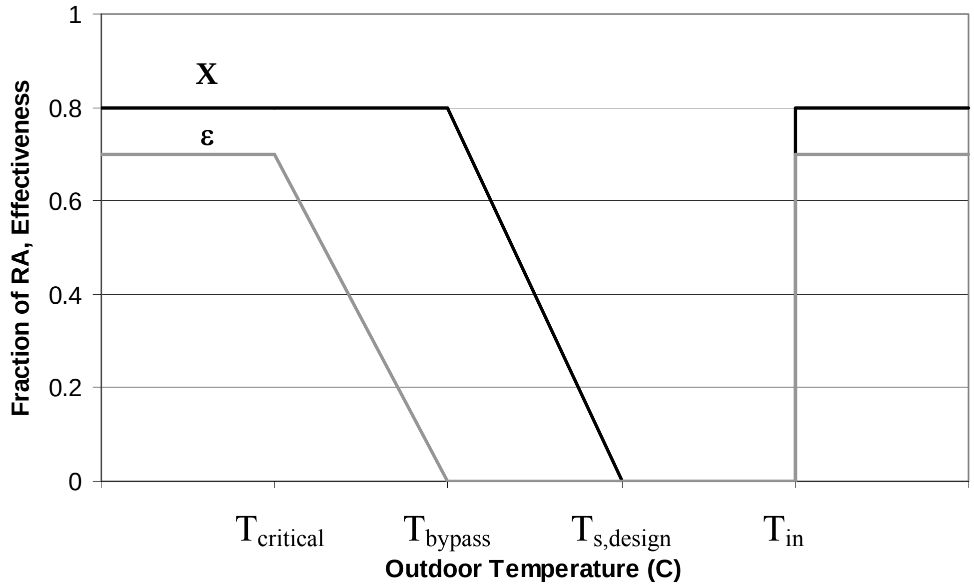


Figure 2.22. Fraction of recirculation air (RA) and energy wheel effectiveness as a function of outdoor temperature.

The schedule for the recirculation air is the same as the economizer described in the base case. The sensible and latent effectiveness of the energy wheel is designed to be 70% initially. This is maintained until the outdoor temperature increases to a certain value, called $T_{critical}$. This value is the temperature of the outdoor air entering the energy wheel that will give the design supply temperature leaving the energy wheel. This is determined from Simonson et al. (2000) as,

$$T_{critical} = \frac{T_{s,design} - (\varepsilon_s + X - X\varepsilon_s)T_{in}}{(1-\varepsilon_s)(1-X)} \tag{2.8}$$

For the constant effectiveness simulation, ε_s will be equal to 0.7 and X will be equal to 0.8 under normal operating conditions ($T_{out} < T_{critical}$ and $T_{out} > T_{in}$).

When the outdoor temperature exceeds $T_{critical}$, the effectiveness needs to be reduced to prevent overheating of the supply air. For a given outdoor temperature, the required effectiveness can be determined by

$$\varepsilon_s = \frac{\left(T_{s,design} - T_{outdoor}\right) - X\left(T_{in} - T_{outdoor}\right)}{\left(1 - X\right)\left(T_{in} - T_{outdoor}\right)}, \tag{2.9}$$

where X is equal to 0.8 for the region $T_{critical} < T_{out} < T_{bypass}$. For the constant variable simulations it is assumed that ε_l equals ε_s. When the outdoor temperature reaches T_{bypass}, the effectiveness reaches zero. At this point all of the air is being bypassed around the energy wheel.

For the variable effectiveness simulations the initial values of ε_s and ε_l, that is the values where $T_{out} < T_{critical}$, are determined from equations (2.3) and (2.4) using the parameters in Table 2.12. As they are calculated from separate equations it can no longer be assumed that ε_l is equal to ε_s for all operating conditions. When $T_{critical} < T_{outdoor} < T_{bypass}$ equation (2.9) is still used to calculate ε_s and ε_l is now calculated from

$$\varepsilon_l = \frac{\varepsilon_{lo}\left(T_{bypass} - T_{outdoor}\right)}{\left(T_{bypass} - T_{critical}\right)}, \tag{2.10}$$

where ε_{lo} is the initial value of ε_l, calculated from equation (2.4) using the parameters in Table 2.12.

CHAPTER 3

SIMULATION DESCRIPTIONS

3.1 SEARCH FOR A COMPUTER SIMULATION PROGRAM

The search for a computer simulation program began with the Building Energy Software Tools Directory (United States Department of Energy, 2003), a website listing 265 different energy-related software tools. From this website a list was made of all the software packages that might have the desired capabilities, including total energy recovery, moisture storage and the ability to calculate indoor temperature and relative humidity. Also, it is beneficial to be able to access the source code of the program. Information was gathered on each of these programs via a journal article search, an internet search and finally communication with the program manufacturers. This information was gathered for 11 computer programs: BLAST, BSim, CHVAC, DOE-2, EnergyPlus, HAP, IDA, ICE, NewQUICK, RIUSKA, TRNSYS and UMIDUS. From this investigation four programs were chosen as the most suitable: DOE, EnergyPlus, BSim and TRNSYS. More research was done and demonstration copies were obtained for BSim and TRNSYS. From this further research it was determined that TRNSYS would be the best program for this thesis. Appendix B contains more details on the search for a computer simulation package.

3.2 SIMULATION OVERVIEW

The TRNSYS computer program is actually a package of several programs that work together to create models of buildings and HVAC systems and simulate their performance. TRNSYS is a FORTRAN based program and is not overly easy to use on its own. For this reason it comes with a program called IISiBat (The Intelligent Interface for the Simulation of Buildings) which is a user friendly interface for creating a model of the HVAC system. Before a model of the HVAC system can be created in IISiBat though, the building must first be created. This is done using the SimCAD and PREBID programs. The SimCAD program is used to create the geometry of the building and to specify the different zones within the building. This building is then input into the PREBID program where all the internal details are specified, including wall and window construction, ventilation, infiltration and occupancy and equipment schedules. Once the building is finished it is then input as a component in the IISiBat workspace. A schematic of the information flow between the programs is shown in Figure 3.1. A detailed description of each program is given in Appendix C. The IISiBat simulation file is given in Appendix D, showing the inputs and outputs of each component in the IISiBat workspace.

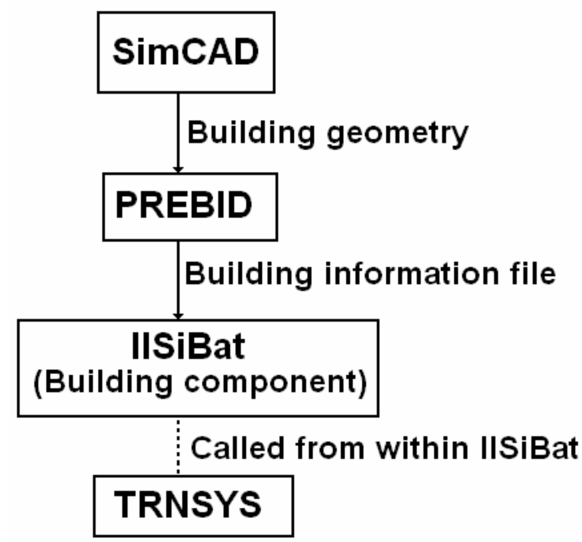


Figure 3.1. Schematic of the information flow between the programs in the TRNSYS package.

The TRNSYS program has been widely used since 1975. According to the manufacturer's website (Solar Energy Laboratory, 2005), the TRNSYS program does not need to be validated. The program is merely a calculation tool that uses standard numerical techniques to calculate the equations of each component used in the IISiBat workspace. The components (heater, cooling unit, energy wheel, etc.) themselves must be validated to ensure they work properly. The main component, the building model is compliant with ASHRAE Standard 140 (2001) - Standard Method of Test for Evaluation of Building Energy Analysis Computer Programs.

3.3 WEATHER DATA

The simulations are performed in four different cities. These are: Saskatoon, Saskatchewan; Vancouver, British Columbia; Phoenix, Arizona; and Tampa, Florida. The global position and elevation of each of these cities can be found in Table 3.1. This information is taken from the weather data files that are used in the TRNSYS program. The weather files used in this thesis are in the TMY2 format and are obtained from the United States Department of Energy website (2006). The weather data is given for each hour throughout the year. When the simulation is run at intervals of less than one hour (as it is here) the data in the weather file is interpolated to give data at each interval.

Table 3.1. Location and elevation of the four cities.

	Saskatoon	Vancouver	Phoenix	Tampa
Latitude	52.17°	49.18°	33.43°	27.97°
Longitude	106.68° W	123.17° W	112.02° W	82.53° W
Elevation	504m	2m	339m	3m

The four cities are chosen based on their typical weather patterns. Saskatoon was chosen for its cool, dry climate, Vancouver for its mild, humid climate, Phoenix for its hot, dry climate and Tampa for its hot, humid climate. The cumulative temperature, humidity ratio and enthalpy distributions for each of these cities are given in Figures 3.2, 3.3 and 3.4, respectively.

From Figure 3.2 it can be seen that each of the four cities has a different temperature profile. In Saskatoon the temperature gets quite cold, and most of the year (8000 hours or 91%) the temperature is below 20°C (68°F). In the other three cities it does not get nearly as cold, with most of the time being above 0°C (32°F). Vancouver has a moderately flat profile indicating that the temperature doesn't vary much throughout the year. Phoenix has the hottest climate, reaching values of almost 50°C (122°F). Tampa also has a warm climate ranging between 0°C and 36°C (97°F). The slope of the cumulative temperature curve in Tampa is large at first and the temperature reaches

20°C quickly, but it then flattens out in the middle stretch. Another slight increase occurs at the end of the profile and the temperature reaches about 35°C (95°F). The maximum and minimum values for temperature, humidity ratio and enthalpy, taken from the weather files are found in Table 3.2.

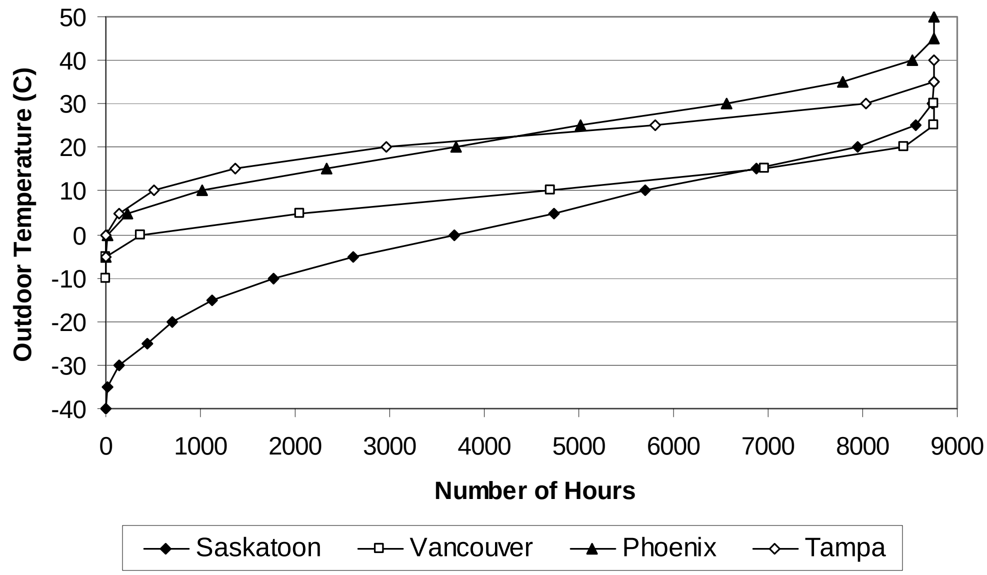


Figure 3.2. Cumulative outdoor temperature distributions for Saskatoon, Vancouver, Phoenix and Tampa.

The cumulative outdoor humidity ratio is shown in Figure 3.3 for each city. Saskatoon, Phoenix and Vancouver have very similar profiles, showing many hours at the lower humidity levels and gradually increasing up to values around 14 g/kg. Tampa, however, shows only a few hours at the lower humidity levels and many hours at higher humidity levels. Tampa peaks around 20 g/kg. The maximum and minimum humidity in each city levels are found in Table 3.2.

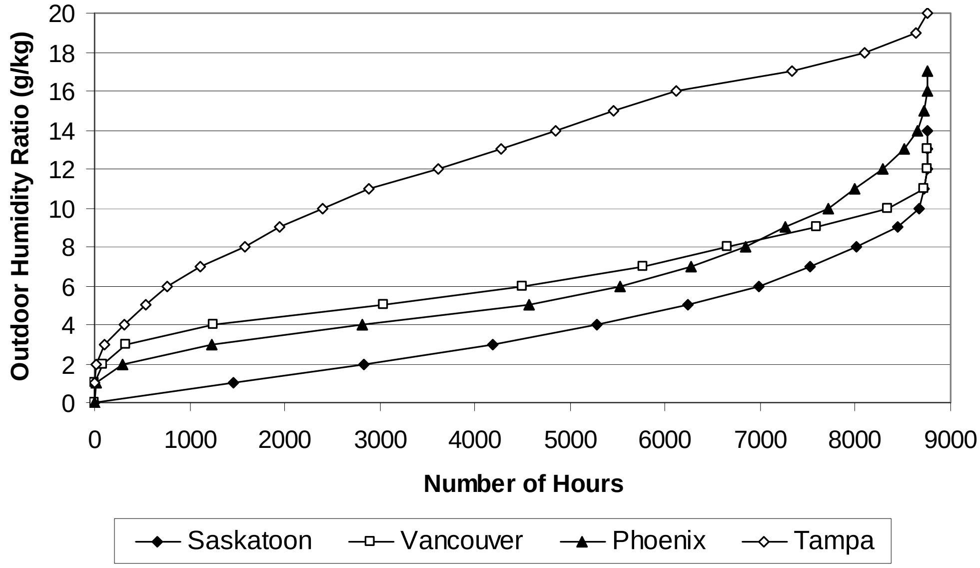


Figure 3.3. Cumulative outdoor humidity ratio distributions for Saskatoon, Vancouver, Phoenix and Tampa.

The cumulative outdoor enthalpy distribution for each city is shown in Figure 3.4. Saskatoon has the lowest enthalpies, because of its low temperatures and humidity levels. Vancouver shows the smallest difference in its range of enthalpies, because of its mild climate. Phoenix is slightly higher, because of its warmer temperatures. Tampa has the highest enthalpies, because of its high temperatures and humidity levels.

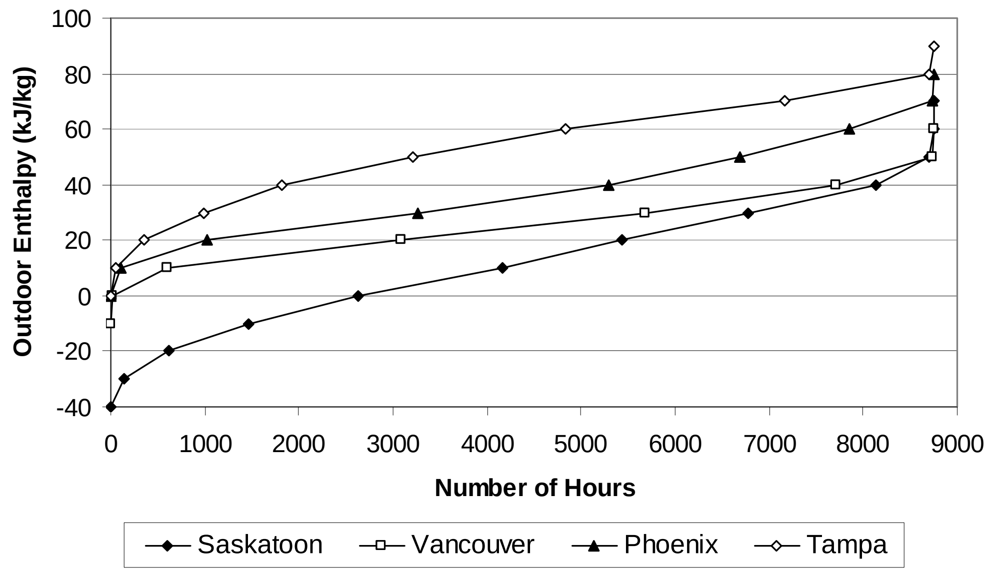


Figure 3.4. Cumulative outdoor enthalpy distributions for Saskatoon, Vancouver, Phoenix and Tampa.

Table 3.2. Maximum and minimum temperatures, humidity ratios and enthalpies for each of the four cities.

	Temperature (°C)		Relative Humidity (%)		Humidity Ratio (g/kg)		Enthalpy (kJ/kg)	
	min	max	min	max	min	max	min	max
Saskatoon	-37.9	34.4	15.3	100.0	0.1	13.6	-37.7	62.5
Phoenix	-2.8	46.1	3.2	99.7	0.4	16.4	0.9	74.4
Vancouver	-7.0	25.6	21.9	100.0	1.0	12.3	-4.0	53.0
Tampa	0.0	35.6	19.0	100.0	1.7	20.5	5.8	83.1

3.4 PRELIMINARY RESULTS

In order to show that the results obtained by the computer simulations are reasonable, the temperature, humidity ratio and relative humidity are presented at several locations within the HVAC system. The simulations are all run with a time step of 2.4 min so there are 25 time steps in every hour. The hourly average outdoor, return, mixed air, supply and space conditions are presented in Figure 3.5 on January 1st at 12:00 in Saskatoon for the office building.

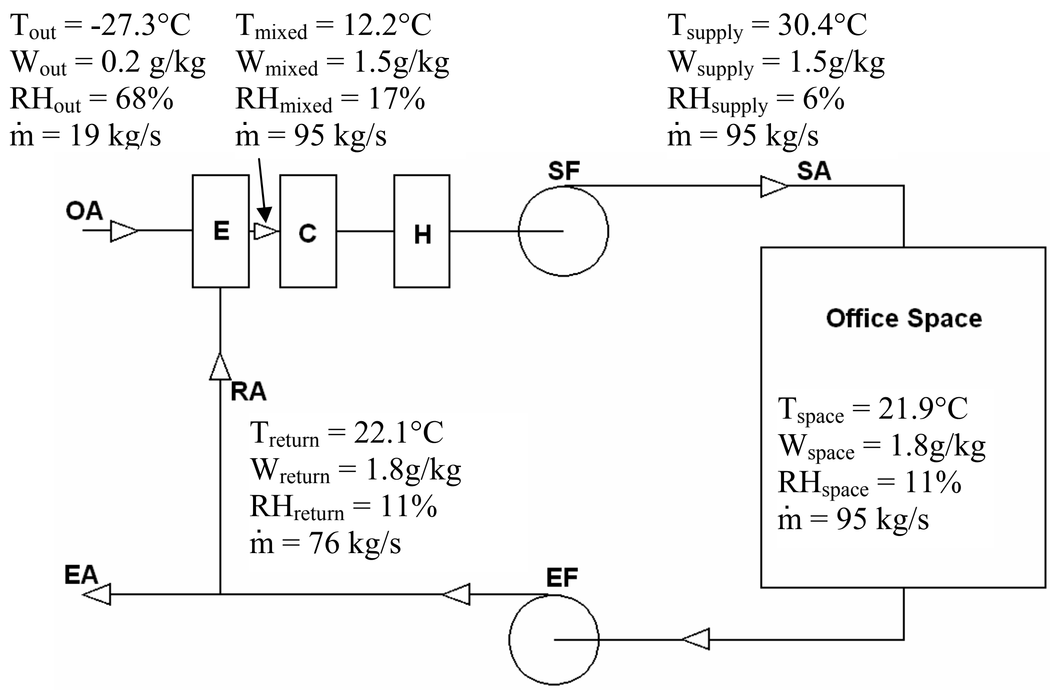


Figure 3.5. Schematic of the HVAC system with air conditions at each location on January 1st at 12:00 in Saskatoon.

As can be seen in Figure 3.5, the outdoor temperature is very cold, (-27.3°C (-17.14°F)) and dry (W = 0.2 g/kg). The return air is at 22.1°C (71.78°F) with a relative humidity of 11% RH. It can be noticed that this is not the same as the air leaving the space. The small increase in temperature is caused by the exhaust fan. The humidity ratio of the space and the return air are the same, as expected. The small difference in temperature also makes a small difference (0.2%) in the relative humidity, but is not noticeable due to rounding.

The conditions of the mixed air can be determined using adiabatic mixing principles which would give a temperature of 12.2°C (53.96°F) and a humidity ratio of 1.5 g/kg. The values shown in the schematic are determined by the computer simulation and agree with hand calculations. Since it is January, and very cold out, the heater is operating to heat the air before it enters the space. The set point temperature for the heater is 30°C (86°F). It can be seen that the supply temperature is 30.4°C (86.7°F). The increase in temperature between the heater set point and the actual supply temperature is due to heat being added by the supply fan. The humidity ratio is the same as the mixed air, which is expected as the heater performs sensible heating only. The relative humidity has dropped because of the increase in temperature.

The results of the base case office building simulation in Saskatoon are presented in this section for two day periods in the heating and cooling season. These results are from the office spaces in the office building, which make up the majority (88%) of the floor area of the building. These results are an hourly average, that is, the average value over the 25 time steps within each hour. The indoor temperature at each hour over two days in January and July can be seen in Figures 3.6 and 3.8, along with the outdoor temperature and the heating or cooling demand.

The results in Figure 3.6 begin at 6:00 on January 1st when the indoor temperature is 18°C due to the night setback. At 6:00, the thermostat calls for additional heat to heat the building to 22°C. This coupled with the fact that the ventilation system is turned on at 6:00 results in the peak heating consumption from 6:00 to 7:00. The temperature in the space begins to heat up so that it is at a comfortable temperature (21.1°C (70.0°F)) when the occupants of the building arrive at 8:00. The temperature in the space rises slightly between 8:00 and 9:00 because of the additional heat given off by the internal heat sources (occupants, lights, equipment) of the space. The increase of heat due to the internal gains reduces the heating energy during the day. At 9:00 the temperature reaches 21.9°C (71.4°F) and levels off there until 23:00 when the ventilation system is shut off for the evening and the night set back is initiated. The heating energy consumption goes to zero at this point. The building slowly cools during the night and at 4:00 the next morning the heating system comes on to keep the temperature above 18°C (64°F). This process is repeated again the next day. The indoor temperature is very similar on the second day, but the energy consumption is slightly lower because the outdoor temperature is higher.

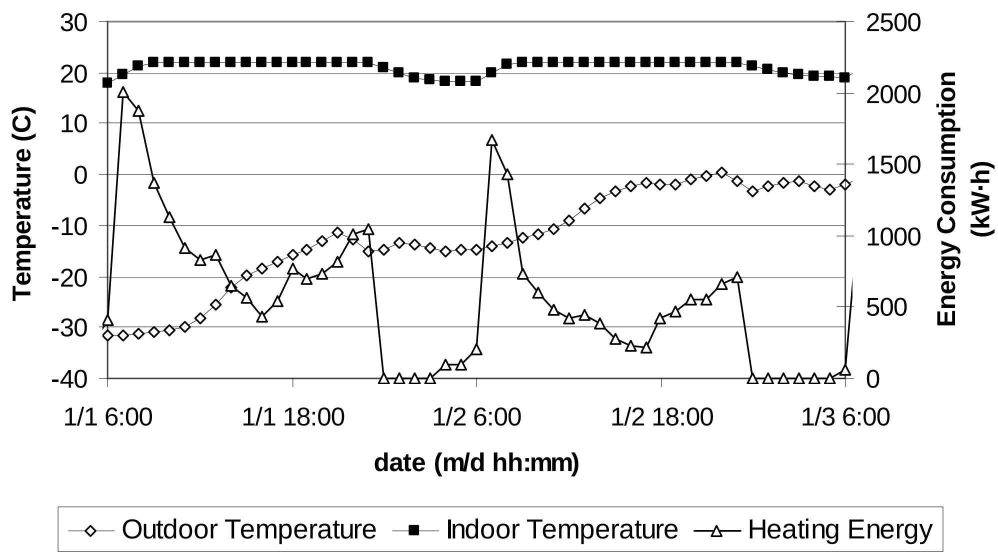


Figure 3.6. Hourly average outdoor temperature, indoor temperature of the office space and heating energy consumption in Saskatoon over a two day period in January.

The relative humidity distribution for the office space during the two day period in January is shown in Figure 3.7, along with the outdoor relative humidity and the heating energy. This profile is more sporadic than the temperature distribution as the relative humidity levels in the building are not controlled. The relative humidity levels depend on the people in the space, as well as the outdoor relative humidity.

The distribution begins at 6:00 when the ventilation system turns on and the indoor space has a relative humidity of about 2% RH. When the people enter the building at 8:00 the relative humidity level begins to increase, reaching 12% RH by 11:00. There is a slight decrease in the relative humidity between 11:00 and 14:00 when some of the occupants leave for lunch. The relative humidity increases again during the afternoon as the occupants return. At 17:00, the majority of the occupants leave the building and the relative humidity level decreases again. By 21:00, all of the occupants have left and the relative humidity levels out at about 7% RH. The process follows the same pattern for the next day but with higher humidity values. These higher values are due to higher outdoor temperatures and consequently higher outdoor humidity ratios on January 2nd. The average outdoor humidity ratio is 0.39 g/kg on January 1st and 1.66 g/kg on January 2nd. The initial relative humidity when the ventilation system turns on is about 2% RH on the first day and increases by 10% RH. On the second day the initial value is 7% RH so an increase of 10% RH would make the relative humidity 17% RH.

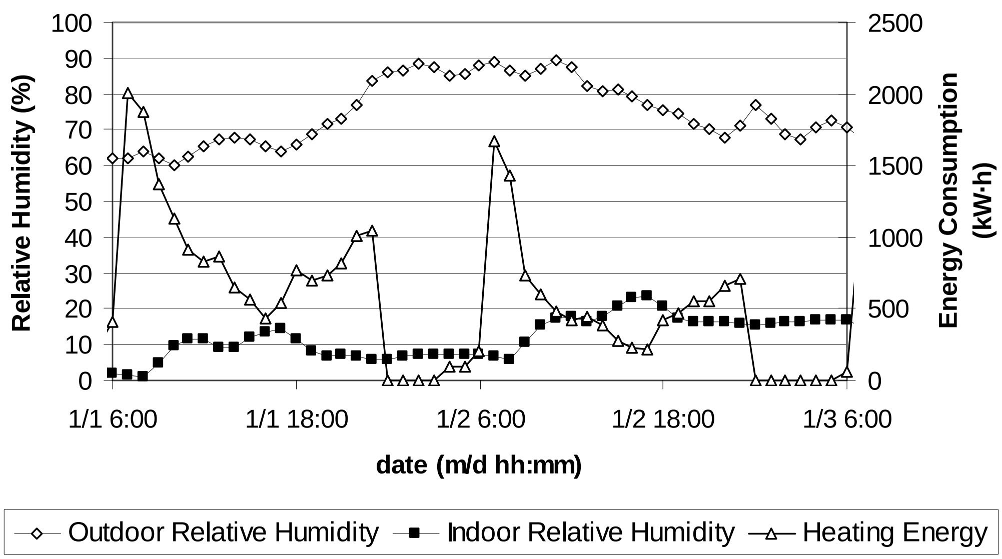


Figure 3.7. Hourly average outdoor relative humidity, indoor relative humidity of the office and heating energy consumption in Saskatoon over a two day period in January.

Figure 3.8 shows the indoor temperature distribution for a two day period in July along with the outdoor temperature and the cooling energy. The indoor temperature is 22°C at 6:00 on July 1st and begins to increase as the occupants enter the building, reaching 24°C at 11:00. At 11:00, the cooling system is turned on to keep the space temperature from exceeding 24°C. The required cooling energy starts out small, but increases as the outdoor temperature increases and radiative gains from previous hours become cooling loads. At 17:00 some of the occupants begin to leave the building so the amount of cooling required to maintain a temperature of 24°C decreases. The temperature in the space begins to decrease at 21:00 when the last occupants leave the building. At this point, the cooling system is turned off and free cooling (the economizer) is used to cool the building. At 23:00, when the ventilation system turns off, the indoor temperature begins to increase again, due to the lighting which is left on at 10% all night. Since the indoor temperature does not reach 24°C during the night the cooling equipment is not turned on again until the next morning.

At 6:00 the next morning (July 2nd), the temperature begins to decrease again. This is because the ventilation system has turned on and is bringing cool outdoor air into the space. As with the day before, the temperature begins to increase when the occupants arrive at 8:00. The outdoor temperatures are higher on July 2nd then on July 1st so the cooling energy required is higher on July 2nd. Since the outdoor temperature is still quite high when the occupants leave the building, no free cooling can be used during this night. The cooling equipment continues to consume a small amount of energy throughout most of the night to maintain the space at 24°C.

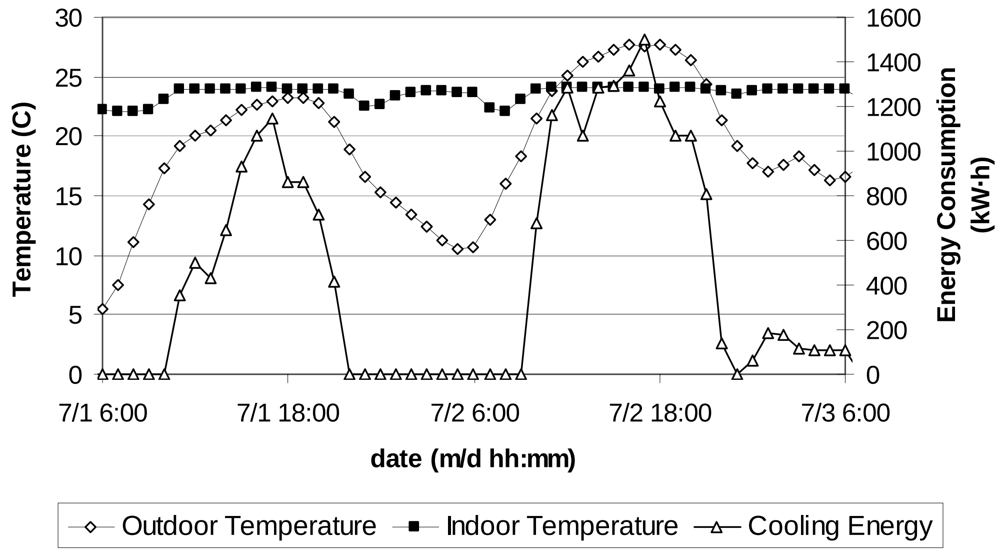


Figure 3.8. Hourly average outdoor temperature, indoor temperature of the office space and heating energy consumption in Saskatoon over a two day period in July.

The outdoor relative humidity, temperature and humidity ratio distributions can be seen in Figure 3.9. The outdoor relative humidity experiences some very rapid changes. These are caused by the changes in the outdoor temperature. During the night, when the outdoor temperature is cool, the relative humidity is quite high. As the temperature begins to rise, the relative humidity decreases, dropping about 50% RH. Both values level off throughout the day. When the temperature begins to decrease again, the relative humidity begins to increase. This pattern is similar on both days. The outdoor humidity ratio remains relatively constant throughout the two days.

Figure 3.10 shows the indoor relative humidity and cooling energy over the two day period in July. The indoor relative humidity is initially at 35% RH at 6:00 on July 1st, but decreases when the ventilation system is turned on because the outdoor humidity ratio is lower than the indoor humidity ratio. The occupants of the building then enter at 8:00 and the relative humidity increases again. At 10:00 the relative humidity begins to decrease rapidly, due to the increase in cooling required, as the cooling process dehumidifies the air. When the occupants leave the building at 21:00 and the amount of cooling required is reduced, the indoor relative humidity begins to increase. At 23:00 the ventilation system is turned off and the relative humidity level within the space remains at a constant value until the next morning. This process is repeated again the next day.

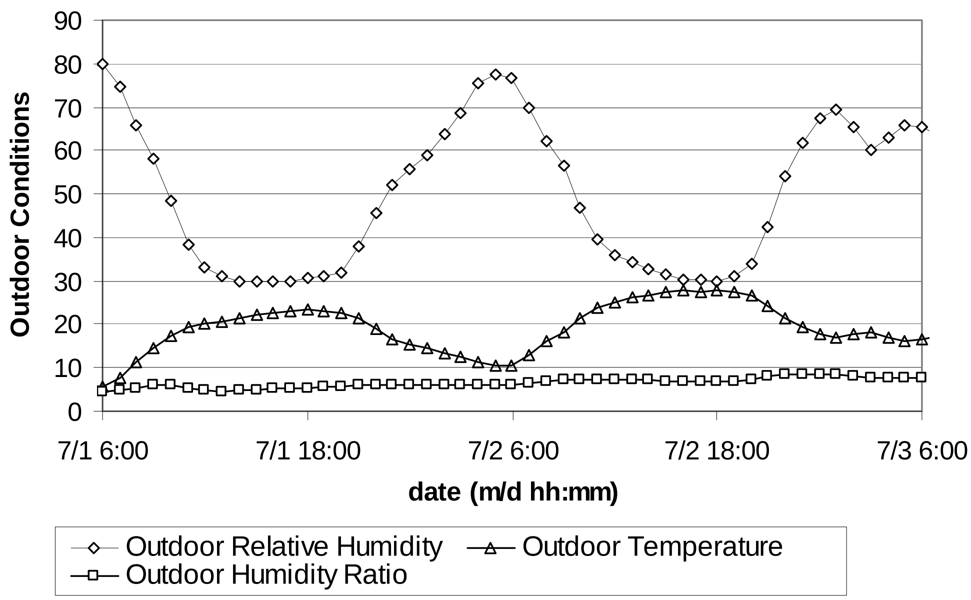


Figure 3.9. Hourly average outdoor relative humidity, temperature and humidity ratio in Saskatoon over a two day period in July.

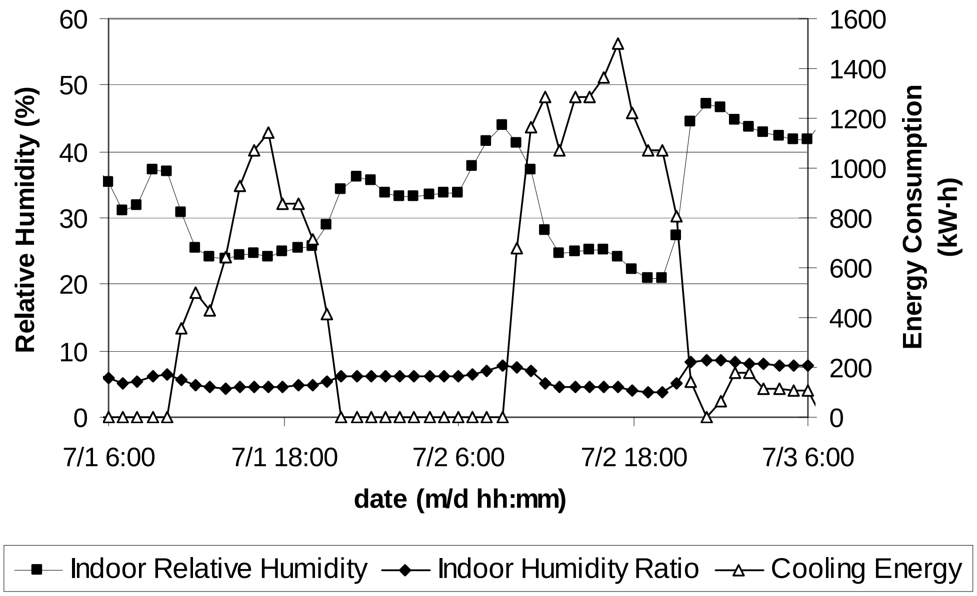


Figure 3.10. Hourly average indoor relative humidity of the office and heating energy consumption in Saskatoon over a two day period in July.

Although the supply temperature is designed to be 14°C (Section 2.3.1), the actual supply temperature will vary based on the outdoor and recirculation air temperatures. The hourly average supply temperature throughout the year can be seen in Figure 3.11. The temperature shown at each point is an average over the 25 time steps that occur during each hour of the simulation. The heater is designed to heat the supply air to 30°C and with the increase in temperature due to the fan, the supply temperature during the heating season is 30.4°C, when the heater is on. Looking at Figure 3.11 it can be seen that the temperature during the heating season is quite often below 30.4°C. This is due to the averaging of the temperature over the 25 time steps within each hour. If the heater is on during all 25 time steps then the average supply temperature will be 30.4°C, but realistically, the heater is only on for a fraction of the total number of time steps, and the average supply temperature is therefore below 30.4°C. The hours that have an average supply temperature close to 30.4°C have a large number of time steps where heating occurs and the hours that have a lower average supply temperature have fewer time steps where heating occurs.

When cooling is required, the supply air temperature varies but is usually between 13°C and 15°C. Figure 3.11 also shows that the supply air temperature varies by more than this two degree range during cooling because, again, the cooling is not on during all 25 time steps. When no heating or cooling is required, the supply temperature varies based on the outdoor and recirculation air temperatures. This range is between 14°C and about 20°C.

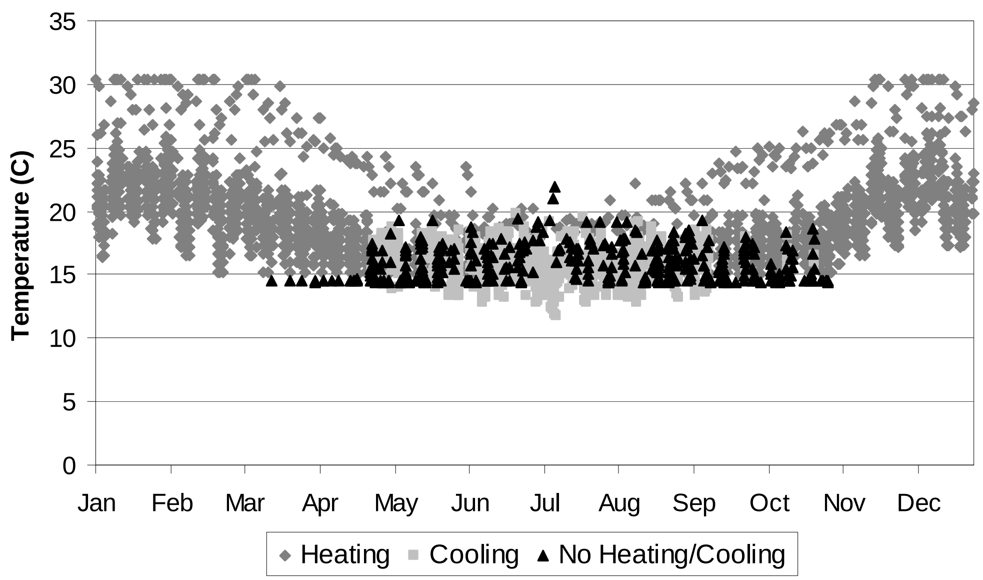


Figure 3.11. Hourly average supply air temperature in Saskatoon during the entire year.

3.5 SUMMARY

Based on the above results it can be seen that all the components in the building simulation are working as expected. For the heating, cooling and yearly periods presented, the temperature and humidity conditions within the HVAC system and occupied spaces are reasonable. The HVAC equipment controls and scheduling appear to be functioning as desired. This serves to verify that the components and the equations they are based on are correct.

CHAPTER 4

INDOOR AIR QUALITY (IAQ) RESULTS

This chapter will discuss the results of the simulations pertaining to the improvement of IAQ, relative humidity levels and the percentage of people dissatisfied with the space conditions. The base case simulation (i.e., no energy wheel in the HVAC system) will be compared against the energy wheel simulation (i.e., with an energy wheel in the HVAC system). As described in Chapter 2, both a constant and variable effectiveness will be considered for the energy wheel.

4.1 EFFECT OF A VARIABLE EFFECTIVENESS ENERGY WHEEL - IAQ

Before looking at the results of the base case, the two energy wheel simulations need be compared to each other, to see the effect that the variable effectiveness has on the results. The indoor space temperature in Saskatoon resulting from the energy wheel simulation with constant effectiveness is shown in Figure 4.1. This is the temperature in the office space at every hour throughout the year. The simulations are created such that the indoor temperature for each case (base, energy wheel with constant effectiveness and energy wheel with variable effectiveness) will be the same as the other cases at any given hour during the year, for the same city. Therefore the difference in indoor temperature between the cases should be minimal. During the winter the temperature varies between 18°C and 22°C which are the set points for heating. During the summer the temperature varies between 22°C and 24°C, the set point for cooling.

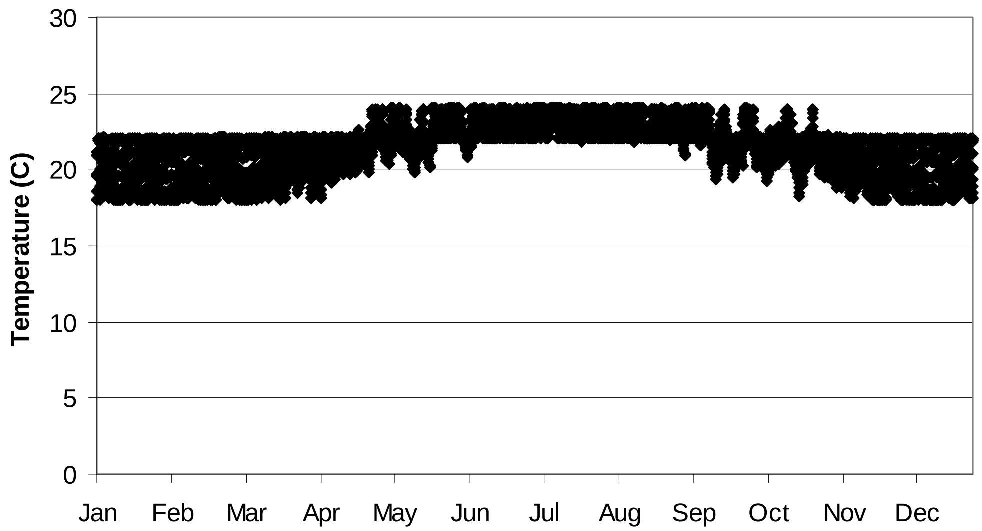


Figure 4.1. Indoor temperature in the office building using an energy wheel with constant effectiveness in Saskatoon.

To see the difference in the results between the constant and variable effectiveness cases a frequency plot is shown in Figure 4.2. To create this frequency plot, the difference in the temperatures is determined by

$$\Delta T = T_{constant} - T_{variable}, \tag{4.1}$$

where $T_{constant}$ is the indoor temperature from the constant effectiveness simulation and $T_{variable}$ is the indoor temperature from the variable effectiveness simulation. The difference in temperature will be positive if the temperature in the constant effectiveness case is larger and it will be negative if the temperature in the variable effectiveness case is larger. The difference in temperature between the two simulations in not expected to be large because the temperature in the space is held constant between the two simulations. Figure 4.2 shows that for over 5000 hours (57% of the year) ΔT is zero, as expected. The maximum and minimum temperature differences are around 0.02°C and -0.02°C, which shows that the variable effectiveness has an insignificant effect on the indoor temperature compared to the assumption of a constant effectiveness for the energy wheel.

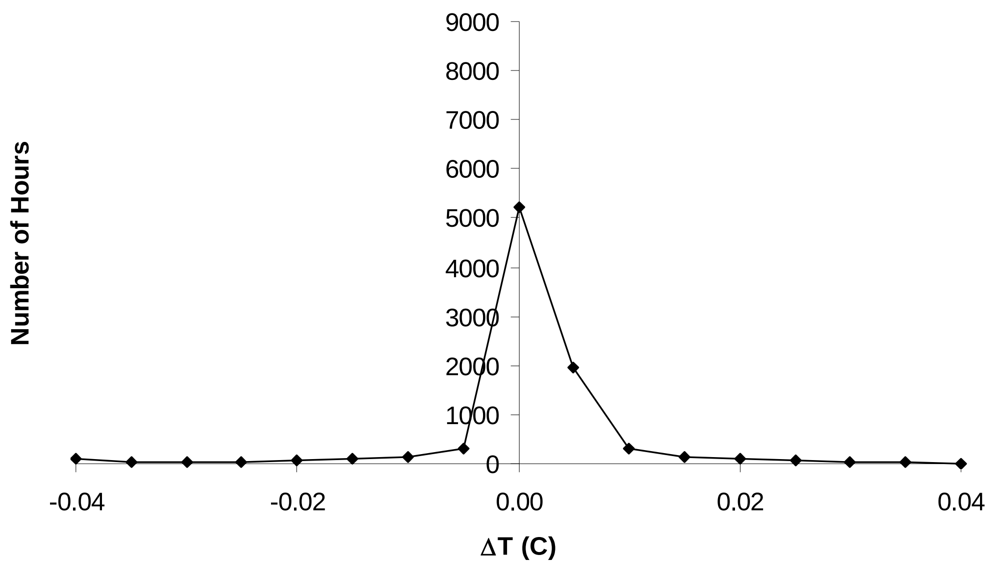


Figure 4.2. Frequency distribution of the difference between the indoor temperature calculated using a constant and variable effectiveness for the energy wheel in Saskatoon.

Figure 4.3 shows the indoor relative humidity distribution for the simulation with a constant effectiveness energy wheel. Again, these results are an hourly average for the office space for the entire year in Saskatoon.

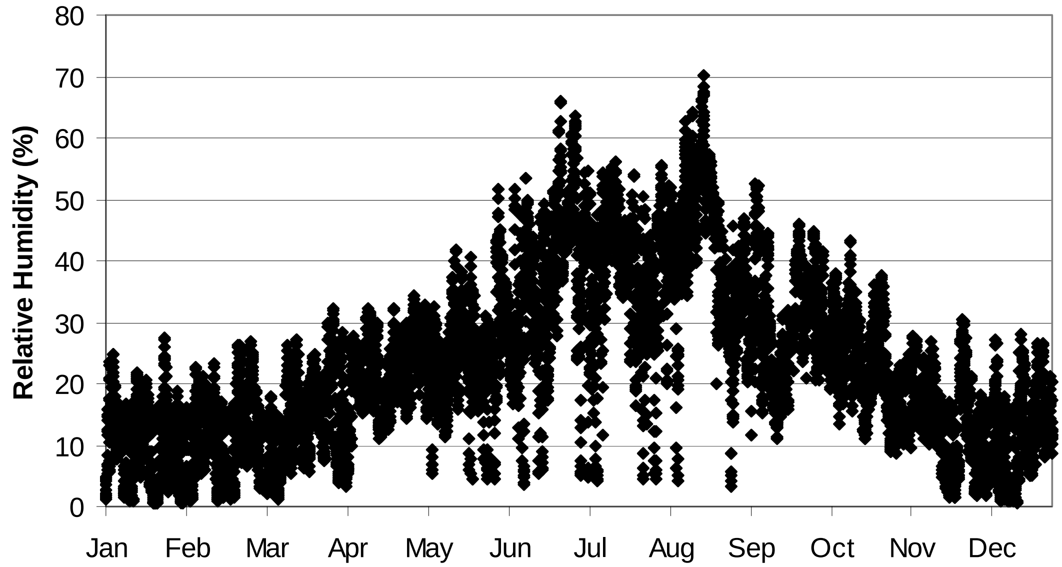


Figure 4.3. Indoor relative humidity in the office building using an energy wheel with constant effectiveness in Saskatoon.

To show the difference in relative humidity between the constant and variable effectiveness cases, a frequency plot is shown in Figure 4.4. The difference in relative humidity is calculated by

$$\Delta RH = RH_{constant} - RH_{variable}, \tag{4.2}$$

where $RH_{constant}$ is the indoor relative humidity for the energy wheel with a constant effectiveness and $RH_{variable}$ is the indoor relative humidity for the energy wheel with a variable effectiveness. Again, it is seen that the difference between the relative humidity calculated with a constant and variable effectiveness is zero for a large portion (55%) of the year. The maximum difference is around 0.5% RH and the minimum around -0.25% RH. The results shown here are from Saskatoon, but similar results occur in all four cities. It can be concluded that for these particular simulations the variable effectiveness has no impact on the indoor temperature and relative humidity. Therefore the remaining simulations in this chapter that include an energy wheel will assume a constant effectiveness for the energy wheel.

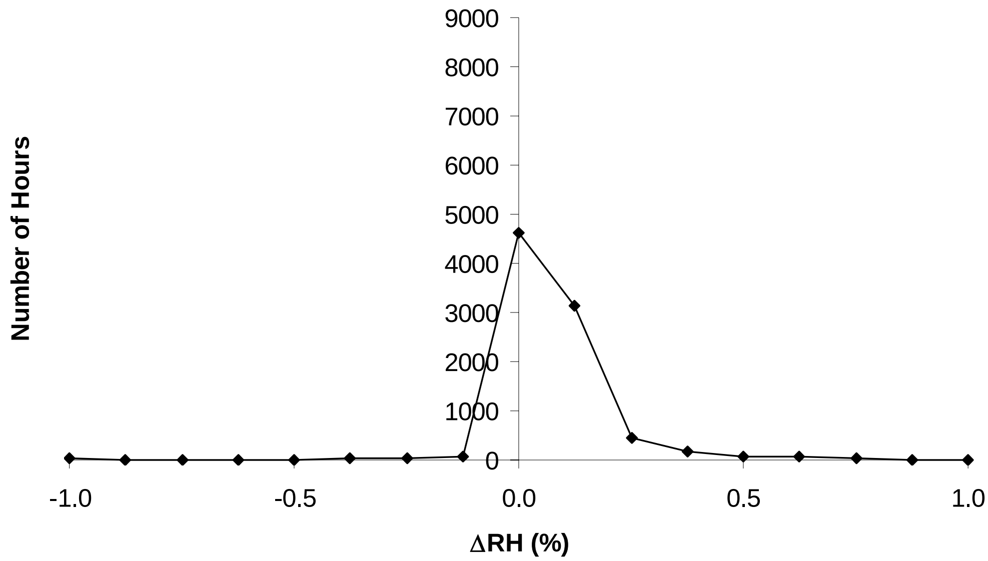


Figure 4.4. Frequency distribution of the difference between the indoor relative humidity calculated using a constant and variable effectiveness for the energy wheel.

4.2 IAQ RESULTS FOR THE OFFICE BUILDING

4.2.1 IAQ in the Office Building in Tampa

The objective of this section is to determine whether the relative humidity within a space can be improved with the use of an energy wheel. Since the relative humidity depends on the temperature within the space, the indoor temperature is kept constant for each of the simulations within the same city, as mentioned above. Figure 4.5 shows the indoor temperature for the base case simulation in Tampa. The temperature is maintained below 24°C and above 20°C throughout the year.

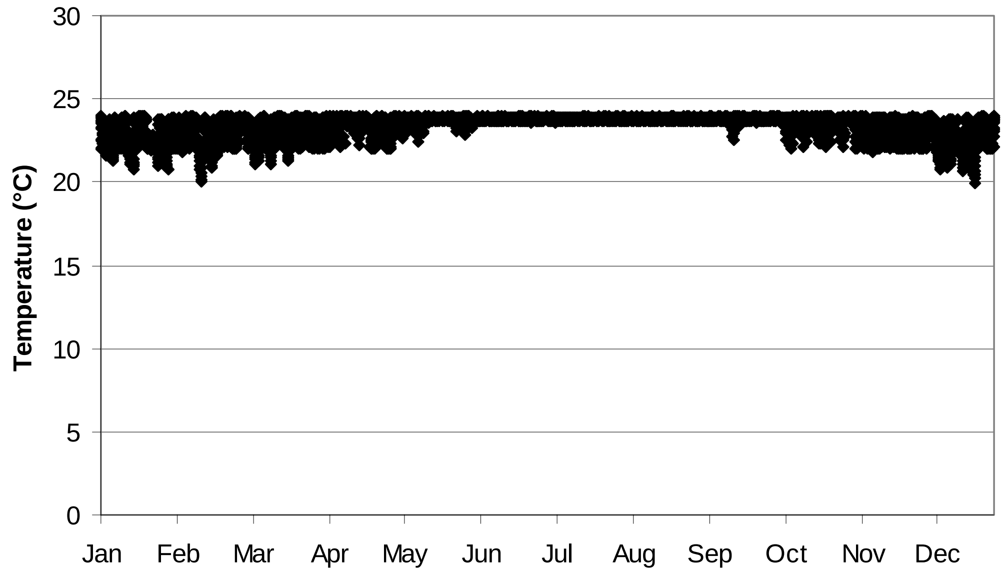


Figure 4.5. Indoor temperature in the office building in Tampa for the base case.

In the above graph it is hard to determine what exactly the temperature profile looks like during the day so Figure 4.6 shows the indoor temperature during occupied hours. This graph shows that during the occupied hours the temperature in the space is always between 22°C and 24°C during the winter months. The variance between these temperatures occurs because the building is cooler (around 22°C) when the occupants enter at 8:00 and gets warmer during the day, reaching 24°C in the afternoon. During the summer months, the temperature is maintained at 24°C throughout the day. Since the comfort and PAQ are only of concern during these occupied hours, the temperature distributions in the remaining cities will only be shown for occupied hours.

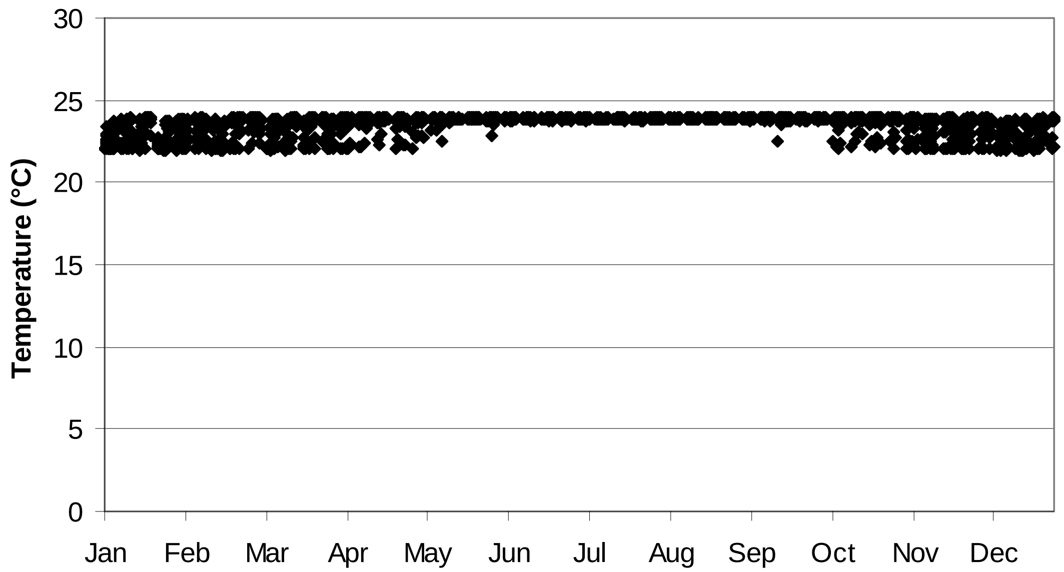


Figure 4.6. Indoor temperature in the office building in Tampa during occupied hours.

Since the objective is to show that the temperature distribution is constant for the base case and the energy wheel case, a frequency plot of the difference in temperature between the two cases is shown in Figure 4.7. The difference in indoor temperature is calculated by

$$\Delta T = T_{base} - T_{energywheel}, \qquad (4.3)$$

where T_{base} is the indoor temperature in the base case and $T_{energywheel}$ is the indoor temperature in the energy wheel case. There are 3393 occupied hours throughout the year. The frequency plot shows that for almost 1000 hours (29% of the occupied hours) the difference in temperature between the two cases (ΔT) is zero. The maximum and minimum values of ΔT are +0.06°C and -0.04°C, with 95% of the ΔT values falling within ±0.02°C. This indicates that the indoor temperature is essentially the same in the two cases. The energy wheel is having no practical impact on the indoor temperature.

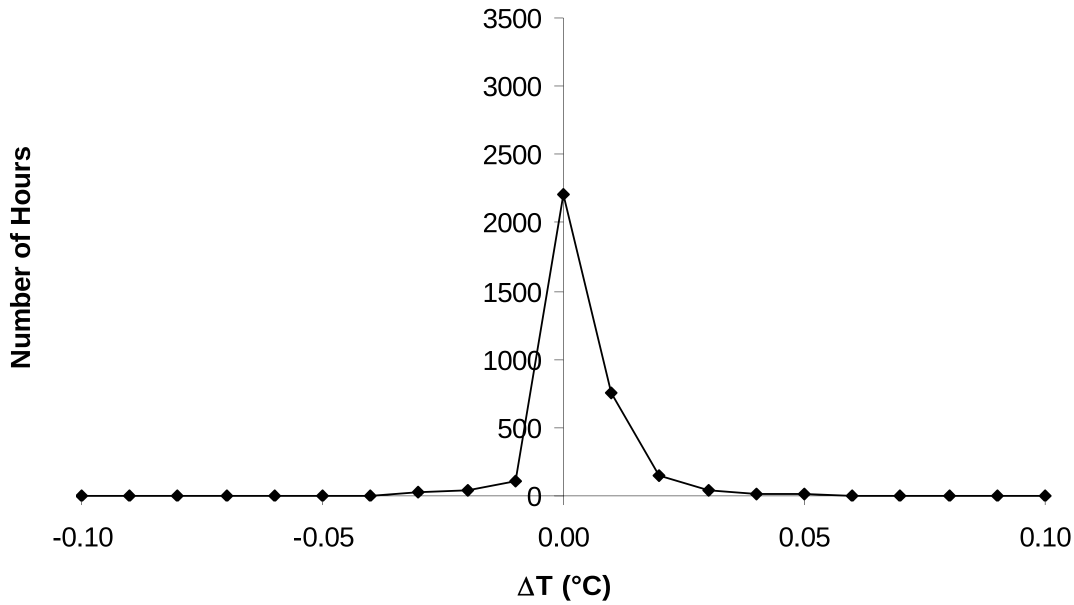


Figure 4.7. Frequency distribution of the difference between the indoor temperature calculated for the base case and the energy wheel case in the office building in Tampa.

The calculated indoor relative humidities in Tampa are shown for every hour throughout the year in Figure 4.8 for the base case. Tampa is quite humid so the indoor relative humidity is quite high as well. The indoor relative humidity exceeds 70% RH during 736 hours (8% of the year) and approaches 90% RH during some hours. It is important to maintain an indoor relative humidity below 70% to avoid moisture problems within the space.

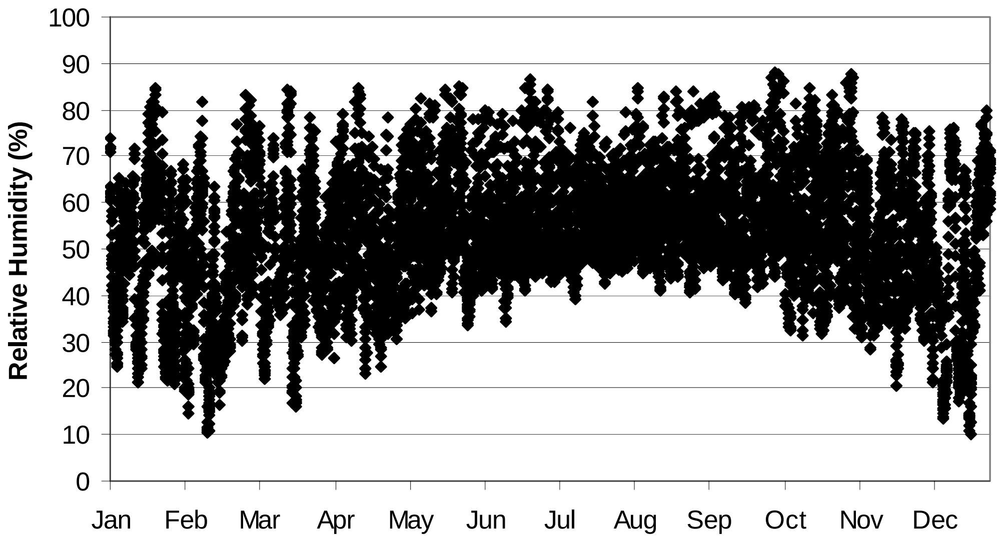


Figure 4.8. Indoor relative humidity in the office building in Tampa for the base case.

Figure 4.9 shows the indoor relative humidity during occupied hours for the base case. It can be seen that the majority of hours where the relative humidity is extremely high are during the unoccupied hours. There are still some hours when the indoor humidity exceeds 70% RH. This occurs during 185 occupied hours (5% of the occupied hours in a year) as opposed to the 736 hours throughout the whole year.

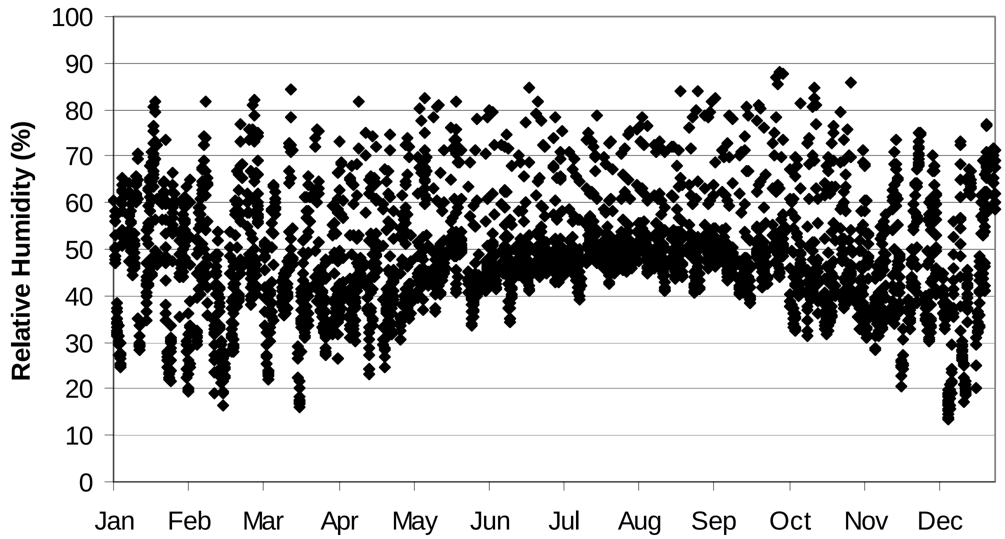


Figure 4.9. Indoor relative humidity in the office building in Tampa during occupied hours for the base case.

The difference in relative humidity (ΔRH) between the base case and the energy wheel case is shown in Figure 4.10 as a frequency plot. The difference in indoor relative humidity is calculated by

$$\Delta RH = RH_{base} - RH_{energywheel}, \qquad (4.4)$$

where RH_{base} is the indoor relative humidity for the base case and $RH_{energywheel}$ is the indoor relative humidity for the energy wheel case. ΔRH will have a positive value when the indoor RH is larger in the base case, indicating that the energy wheel is able to reduce the indoor RH. It should be noted that for this frequency plot the frequency is calculated over a range with the frequency being plotted at the center value of that range. For example, in Figure 4.10 the frequency is calculated for the range 0% RH to 2% RH and the value is plotted at 1% RH.

There are about 1300 hours (38% of the year) with -2% RH < ΔRH < 2% RH, indicating that a majority of the hours (59%) have a |ΔRH| > 2%. There are approximately 1000 hours (29%) where the difference is around 10% RH. The maximum difference is about 16% RH. This shows that the energy wheel can make a significant impact on the indoor RH level in a humid climate.

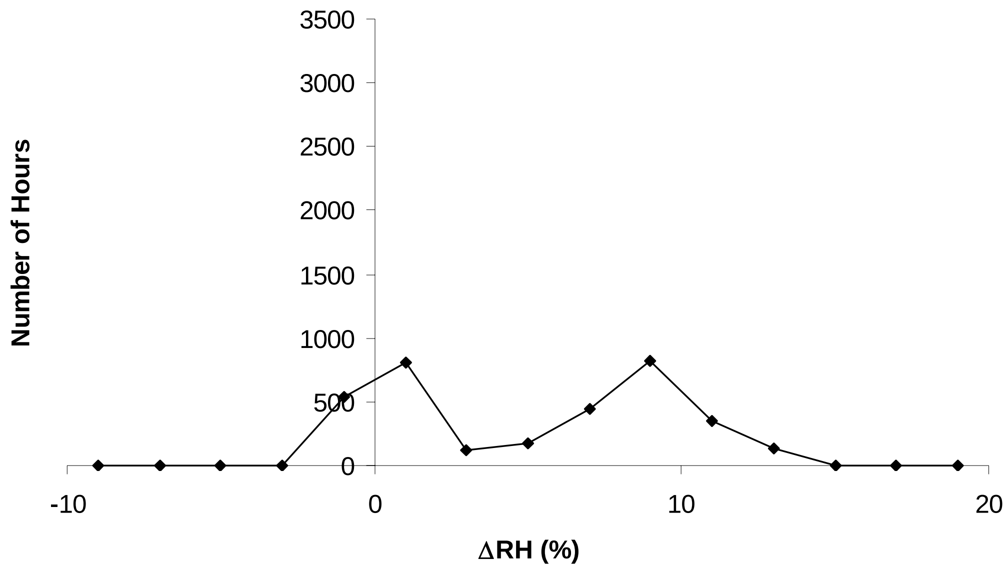


Figure 4.10. Frequency distribution of the difference between the indoor relative humidity calculated for the base case and energy wheel case in the office building in Tampa.

To see how the energy wheel affects the relative humidity during a typical day, an hourly comparison is shown in Figure 4.11, along with the cooling energy for both the base and energy wheel cases. The outdoor temperature, relative humidity and humidity ratio are shown in Figure 4.12 to help explain the changes in RH shown in Figure 4.11. In the base case simulation, the indoor humidity initially decreases from 55% RH to 50% RH between 0:00 and 2:00 due to moisture removed by the cooling coil. At 6:00, the relative humidity begins to increase considerably. This is due to the ventilation system turning on and allowing outdoor air, which is at a higher humidity ratio than the indoor humidity ratio, into the system. At 8:00 when the people enter the building and other heat sources begin generating heat, the cooling required increases, causing a decrease in the relative humidity. Another RH increase is seen at 18:00 when some of the people leave and the demand for sensible cooling is not as high. The cooling demand continues to drop through the evening and the indoor relative humidity continues to increase slowly.

The energy wheel case shows a similar humidity profile as the base case, but with a humidity level that is about 10% RH lower. The impact of the energy wheel is first noticed at 6:00 when the outdoor ventilation system is activated. The energy wheel removes some of the moisture from the outdoor air before it is delivered to the cooling coil and space, reducing the amount of moisture that the supply air adds to the space. The relative humidity profile then continues the same as the base case, but at this lower relative humidity level. During the night, the cooling unit is able to reduce the indoor relative humidity in the base case so that it is similar to the indoor relative humidity in the energy wheel case by 6:00 the next morning. It can be seen in Figure 4.11 that the energy wheel lowers the energy consumption at most hours as expected. The effect of the energy wheel on energy consumption will be analyzed in Chapter 5.

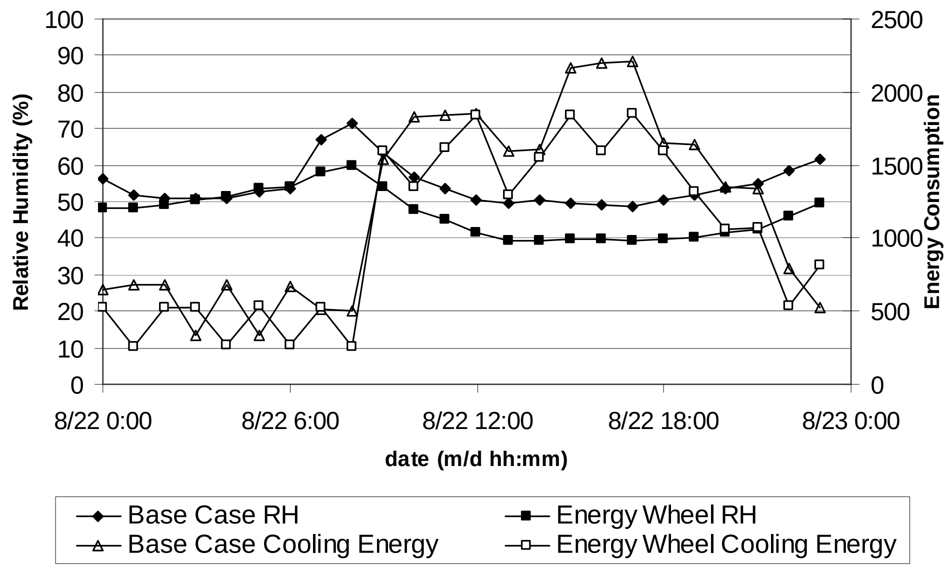


Figure 4.11. Hourly comparison of indoor relative humidity and cooling energy in the office building in Tampa on August 22nd.

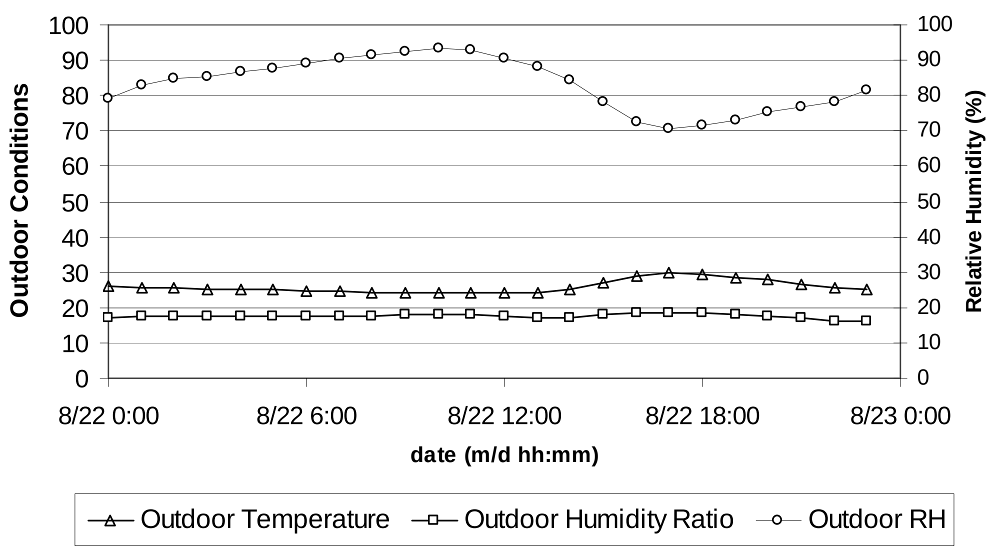


Figure 4.12. Hourly outdoor temperature, relative humidity and humidity ratio in Tampa on August 22nd.

The percentage of people dissatisfied with thermal comfort (PD_{tc}) during occupied hours in Tampa is shown in Figure 4.13. For the base case, PD_{tc} fluctuates between

16% and 19%. There is a consistent decrease of about 2% at every occupied hour when the energy wheel is used. This is not a large change, but still shows that the energy wheel has a positive effect on the conditions in the building during occupation. The energy wheel is not expected to have a large impact on general thermal comfort because general thermal comfort is mostly based on the temperature in the space and the temperature is essentially the same in the two cases.

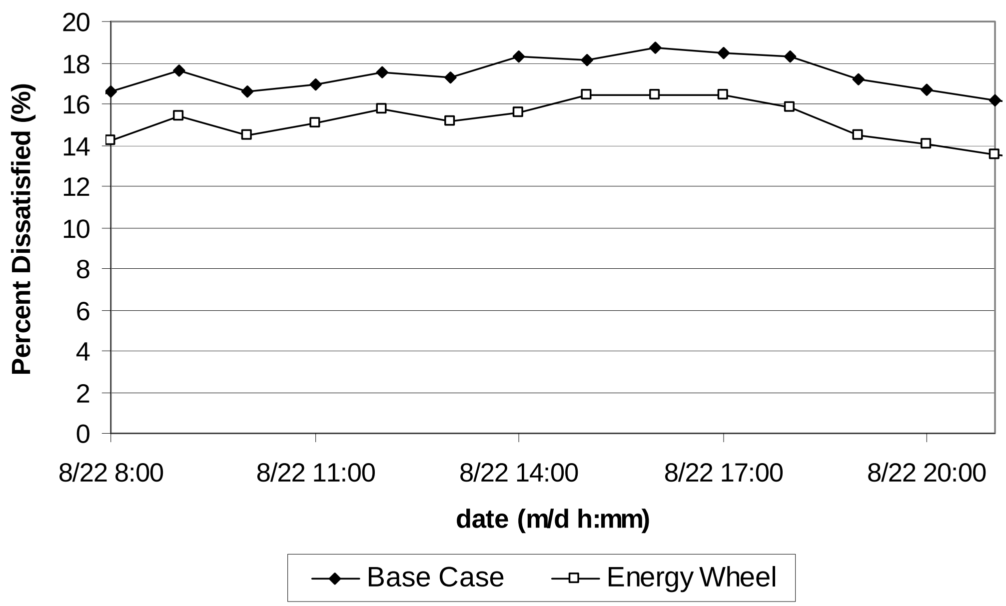


Figure 4.13. Hourly comparison of PD_{tc} in the office building in Tampa on August 22nd during occupied hours.

Figure 4.14 shows the percentage of people that are dissatisfied with perceived air quality (PD_{IAQ}) during occupied hours in Tampa. There is a large decrease in the PD_{IAQ} when the energy wheel is added to the system as compared to the base case. The peak decrease is 22% at 8:00 and the average decrease is about 17%. From these results it can be seen that the addition of an energy wheel into the HVAC system can have a significant impact on the PAQ of a space. This will likely affect the productivity of the office workers and should be considered when conducting a life cycle cost analysis for buildings. This analysis will be left for future work.

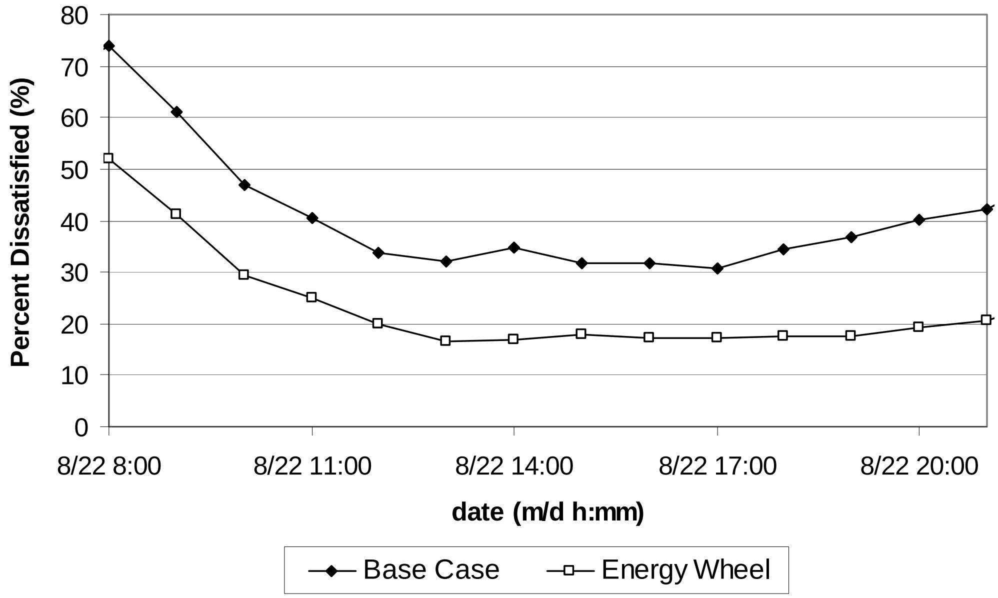


Figure 4.14. Hourly comparison of PD_{IAQ} in the office building in Tampa on August 22nd during occupied hours.

4.2.2 IAQ in the Office Building in Phoenix

The thermal climate in Phoenix is similar to Tampa so the indoor temperature will also be similar. Figure 4.15 shows the indoor air temperature for Phoenix during occupied hours. The temperature in the space is maintained between 22°C and 24°C during the winter months and around 24°C during the summer months.

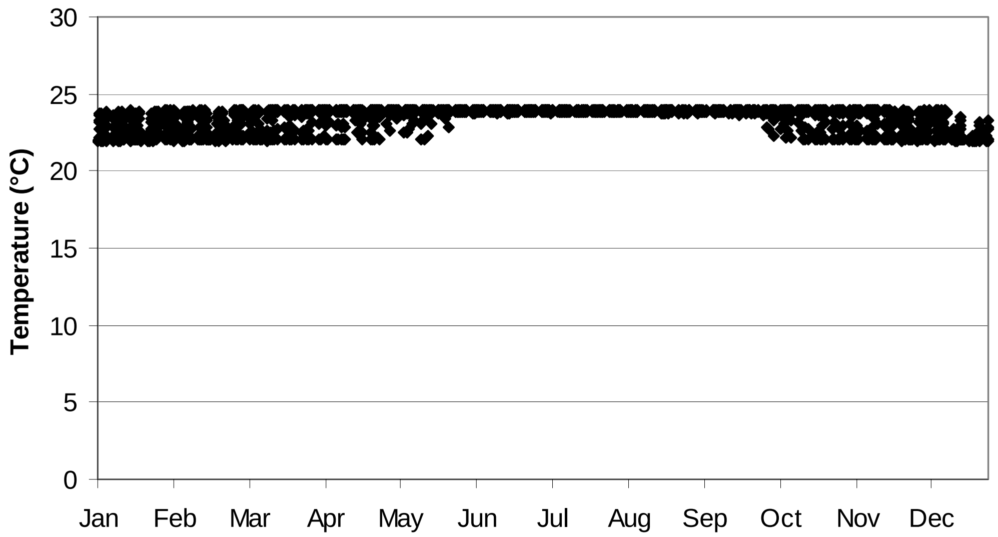


Figure 4.15. Indoor temperature in the office building in Phoenix during occupied hours.

The temperature difference between the base case and the energy wheel case is shown in Figure 4.16. ΔT varies more here than it did in Tampa, but the difference is still quite small. The maximum temperature difference is 0.16°C and the minimum is -0.14°C with 95% of the ΔT values falling within ±0.08°C. This is again an insignificant margin so it is concluded that the indoor temperatures are practically the same whether the HVAC system has an energy wheel or not.

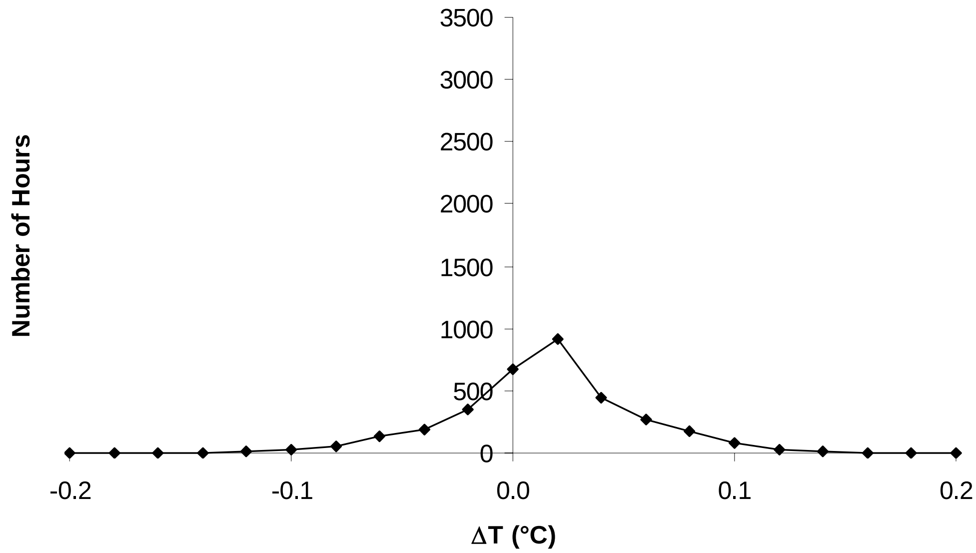


Figure 4.16. Frequency distribution of the difference between the indoor temperature calculated for the base case and energy wheel case in the office building in Phoenix.

The indoor relative humidity during occupied hours in Phoenix, for the base case, is shown in Figure 4.17. As Phoenix is considerably less humid than Tampa, the indoor relative humidity is much lower in Phoenix than in Tampa. The relative humidity in Phoenix is kept below 70% RH at all but two hours so there should be minimal problems due to high indoor humidities in the building.

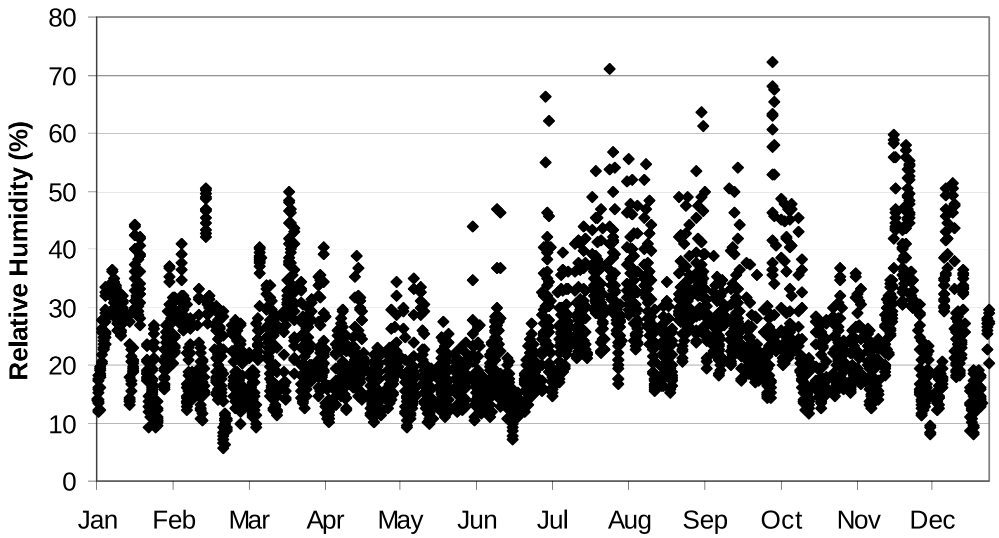


Figure 4.17. Indoor relative humidity in the office building in Phoenix during occupied hours.

The difference in relative humidity between the base case and the energy wheel case during occupied hours is seen in Figure 4.18. The addition of the energy wheel does not affect the indoor relative humidity in Phoenix as much as it does in Tampa, but there are still some differences. There are about 1500 occupied hours (44%) where the relative humidity changes by up to 2% RH in Phoenix. The maximum relative humidity difference is about 11% RH.

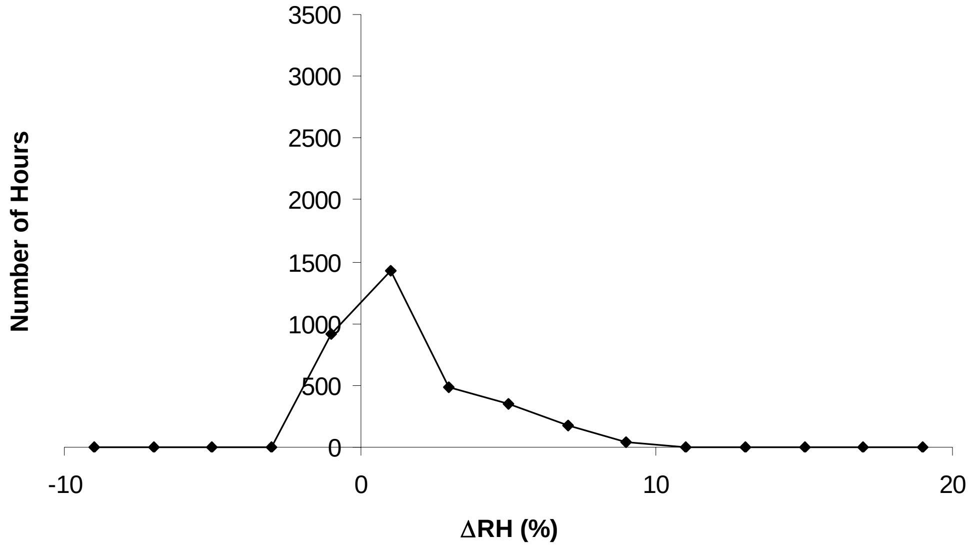


Figure 4.18. Frequency distribution of the difference between the indoor relative humidity calculated for the base case and energy wheel case in the office building in Phoenix.

Figure 4.19 shows the hourly relative humidity in Phoenix on July 3rd. Both the base case and energy wheel case show similar trends to Tampa, with the indoor RH decreasing when the cooling energy increases, and increasing when the outdoor air is introduced at a higher humidity than the indoor air. The cooling equipment increases to remove the additional heat generated by occupants and equipment at 8:00 and begins to decrease at 18:00 when some of the occupants leave. The relative humidity in the energy wheel case follows a similar profile to the base case, except at a lower relative humidity. This is the same situation that happened in Tampa. Between the hours of 6:00 and 8:00 when the cooling load is low and outdoor air is being used, the energy wheel removes a significant amount of moisture from the outdoor air, before it is brought into the space. This reduces the peak relative humidity and subsequently the relative humidity is at a lower value throughout the day as compared to the base case.

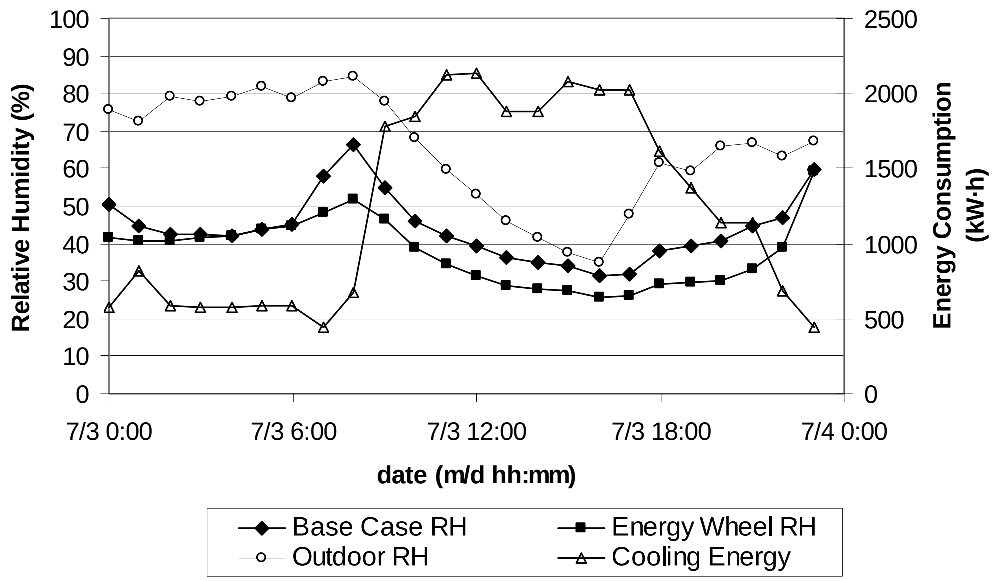


Figure 4.19. Hourly comparison of indoor relative humidity in the office building in Phoenix on July 3rd.

The hourly comparison of PD_{tc} (Figure 4.20) shows a consistent decrease of at least 2% for all occupied hours with the energy wheel, as compared to the base case. This is similar to the results shown in Tampa. The PD_{IAQ} in Phoenix is shown in Figure 4.21 and shows that the energy wheel is able to reduce PD_{IAQ} during all the occupied hours on July 3rd. At certain hours there is a large difference between the base case and the energy wheel case, and at other times there is a smaller difference between the two cases. The peak decrease in PD_{IAQ} is about 30% at 8:00 and the average decrease throughout the occupied hours is 10%.

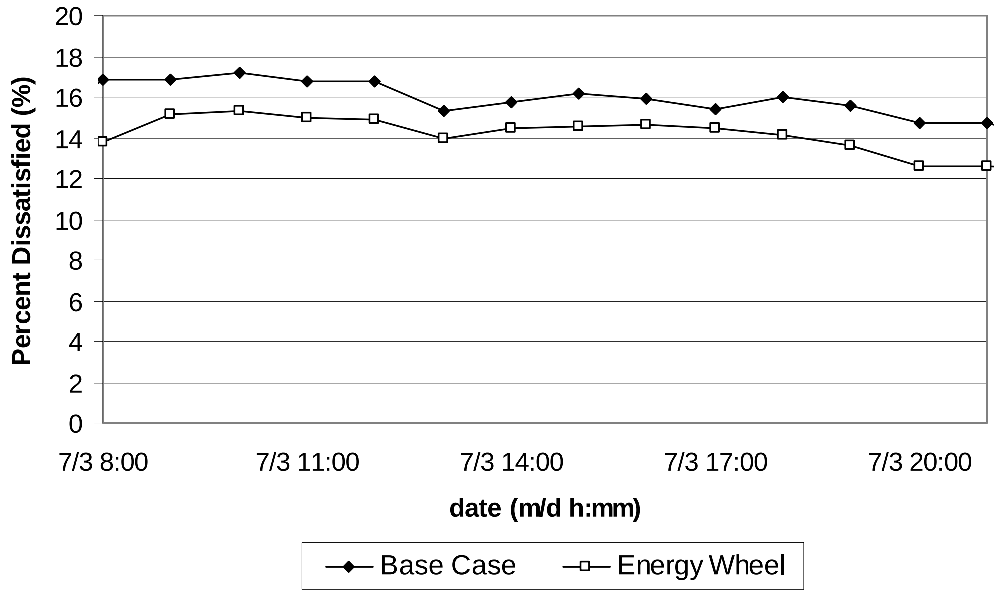


Figure 4.20. Hourly comparison of PD_{tc} in the office building in Phoenix on July 3rd during occupied hours.

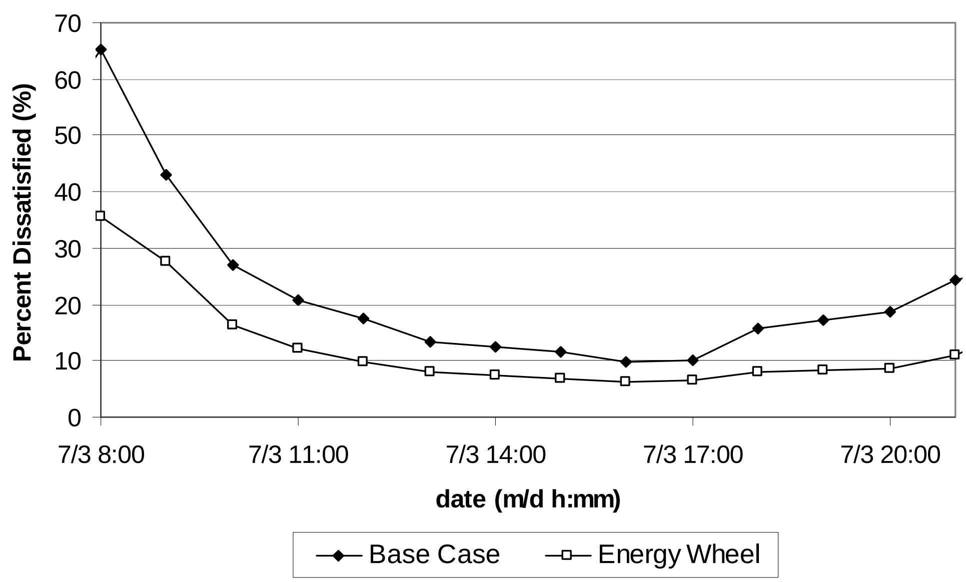


Figure 4.21. Hourly comparison of PD_{IAQ} in the office building in Phoenix on July 3rd during occupied hours.

4.2.3 IAQ in the Office Building in Saskatoon

The city of Saskatoon has a much colder climate than both Tampa and Phoenix. As seen in the temperature distribution for occupied hours below (Figure 4.22), the indoor temperature is maintained at 22°C during the winter months and varies between 22°C and 24°C during the summer months. There are a few times throughout the winter when the heating equipment is not able to maintain the indoor temperature at 22°C. At these times, the outdoor temperature is very cold and the heating equipment does not have the capacity to heat the indoor air to 22°C. This only occurs between 6:00 and 8:00, before the occupants enter the building and only for a few days of the year and therefore is not considered critical.

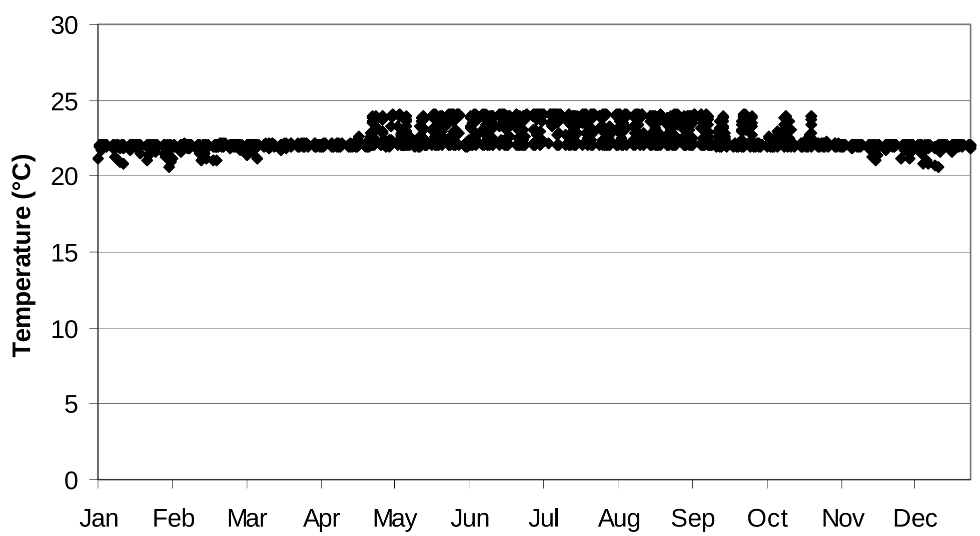


Figure 4.22. Indoor temperature in the office building in Saskatoon during occupied hours.

The indoor temperature difference between the base case and the energy wheel case is shown in Figure 4.23. Once again, the energy wheel has a negligible impact on the indoor temperature and the maximum value of ΔT is 0.1°C and the minimum value is -0.06°C.

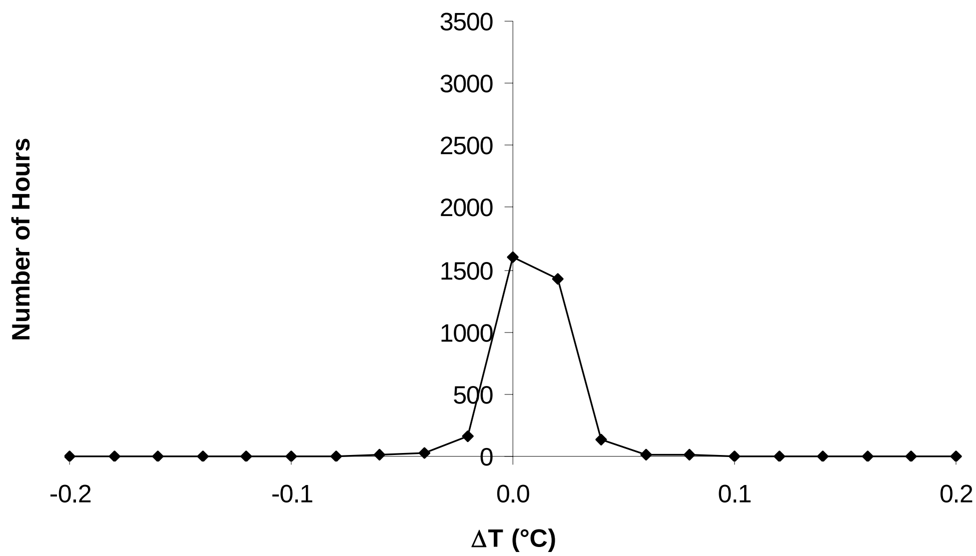


Figure 4.23. Frequency distribution of the difference between the indoor temperature calculated for the base case and energy wheel case in the office building in Saskatoon.

The indoor relative humidity during occupied hours (Figure 4.24) is considerably lower in Saskatoon than in Tampa and Phoenix. This is because of the very dry climate in Saskatoon. Since the relative humidity is so low in the winter months it is favorable to not only decrease the relative humidity in the summer months, but also to increase the relative humidity in the winter months.

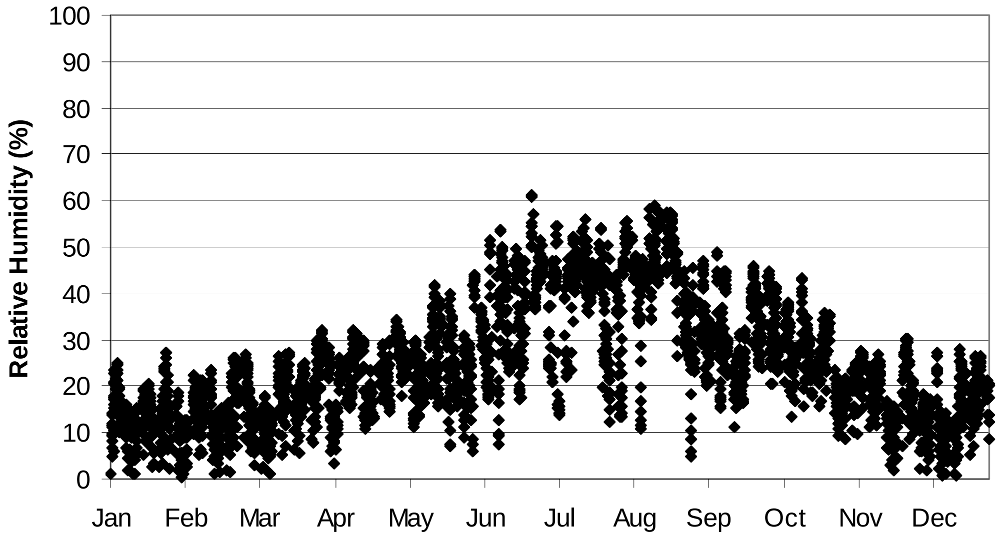


Figure 4.24. Indoor relative humidity in the office building in Saskatoon during occupied hours.

Figure 4.25 shows the difference in relative humidity between the base case and the energy wheel case during occupation. There are about 2000 hours (59% of occupied hours) where the relative humidity difference is 0% RH, indicating that the energy wheel does not have a huge effect on the indoor conditions in Saskatoon. The maximum difference is about 4% RH and the minimum difference is about -4% RH. A relative humidity difference of -4% RH means that the energy wheel increases the relative humidity in the building, which may be desirable during cold, dry weather, but not during humid weather.

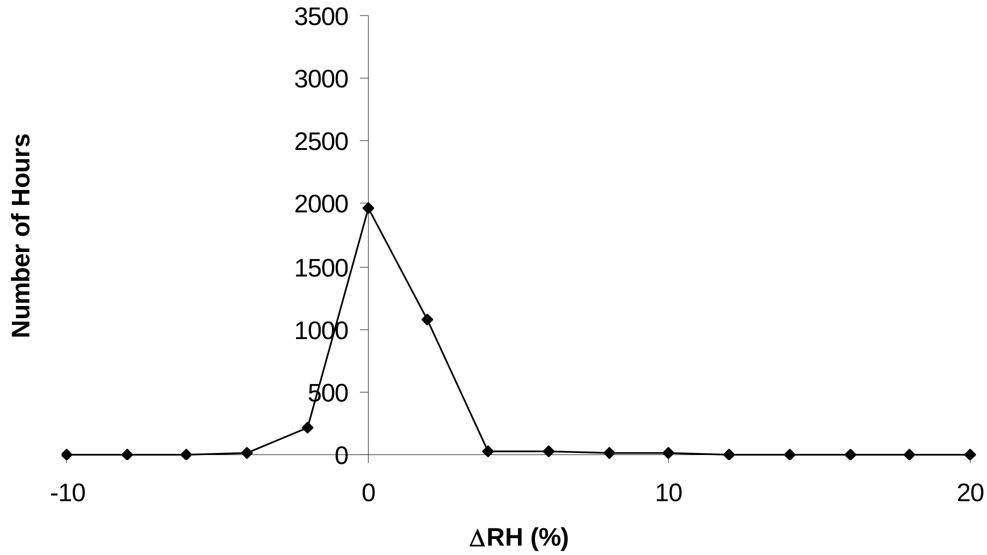


Figure 4.25. Frequency distribution of the difference between the indoor relative humidity calculated for the base case and energy wheel case in the office building in Saskatoon.

Figure 4.26 shows the daily average indoor relative humidity in Saskatoon, during occupied hours. During the summer months, the energy wheel case shows a decrease in the average relative humidity, as compared to the base case. In the winter however, the energy wheel acts to increase the daily average relative humidity level over the base case level. This shows that the energy wheel has a generally positive effect on the indoor relative humidity in both the summer and the winter.

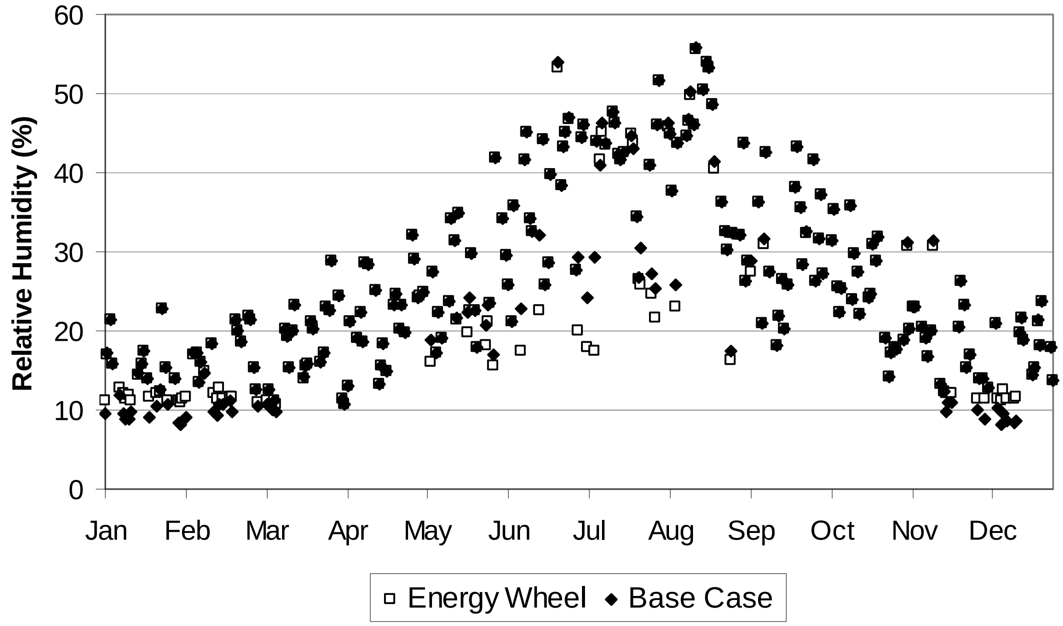


Figure 4.26. Daily average indoor relative humidity in the office building in Saskatoon during occupied hours.

The hourly comparison of the indoor relative humidity in Saskatoon is shown for December 16th and July 8th in Figures 4.27 and 4.28, respectively. The outdoor conditions for July 8th are shown in Figure 4.29. The indoor relative humidity is very low in the winter, around 2% RH for the base case simulation. At 8:00, when the occupants begin to arrive, the relative humidity increases to about 11% RH. The relative humidity then begins to decrease again around 18:00 when the occupants begin to leave and levels out at about 2% RH during the night. The simulation with the energy wheel also shows a very low relative humidity when the occupants enter the building at 8:00. When the indoor humidity begins to increase, however, it increases to about 15% RH with the energy wheel, compared to 10% RH in the base case. The energy wheel results then follow the same profile as in the base case, but about 4% RH higher.

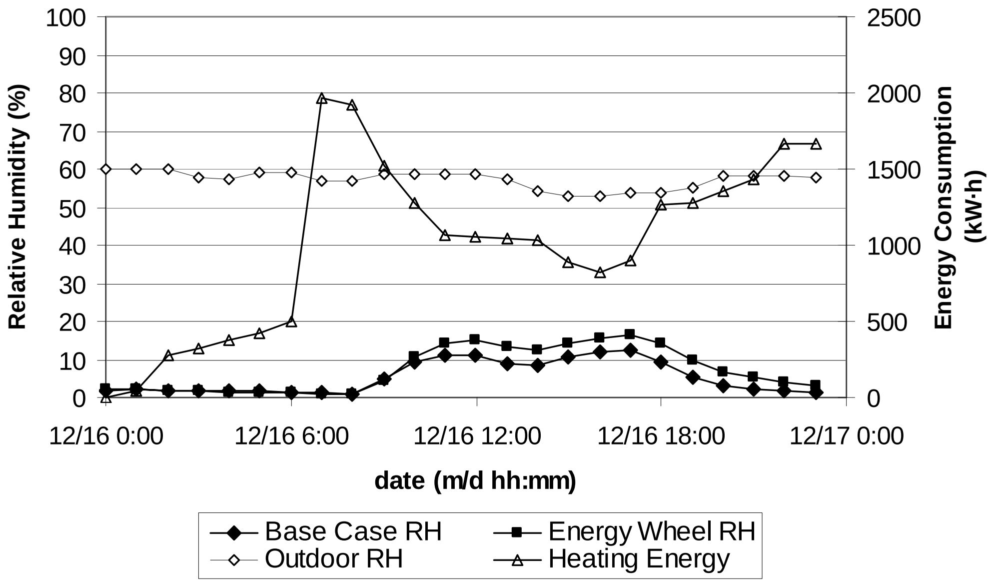


Figure 4.27. Hourly comparison of indoor relative humidity in the office building in Saskatoon on December 16th.

For the base case simulation on July 8th, the indoor humidity starts at about 35% RH, as seen in Figure 4.28. The indoor RH level begins to increase at 6:00 when the ventilation system is turned on because the outdoor air has a slightly higher humidity ratio than the indoor air. Around 10:00, the indoor RH begins to decrease, due to the large cooling loads which would be removing a lot of moisture from the supply air. The indoor RH begins to increase again at 21:00 when the occupants have left and other internal heat sources are off, and the required cooling is lower.

The indoor relative humidity in the energy wheel case follows a similar profile to the base case, but the decrease in relative humidity during the day is considerably larger. This drop is a result of the outdoor humidity ratio. Since the outdoor humidity ratio is still higher than the space humidity ratio, moisture is being transferred from the outdoor air stream to the exhaust air stream in the energy wheel. The relative humidity in the energy wheel case continues to drop until it reaches about 5% RH. Although it is beneficial to reduce the relative humidity in most cases, a minimum value should be maintained to prevent problems due to the air being too dry.

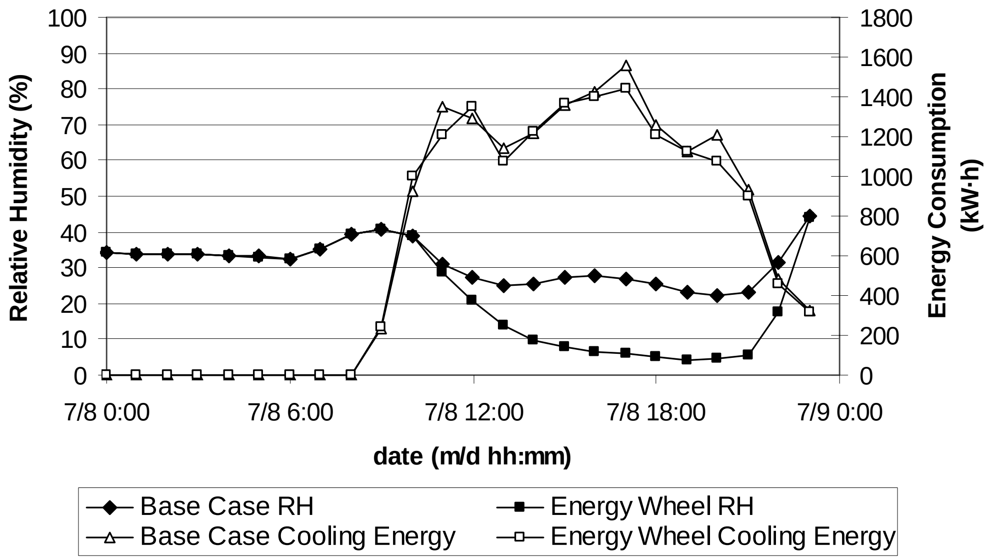


Figure 4.28. Hourly comparison of indoor relative humidity in the office building in Saskatoon on July 8th.

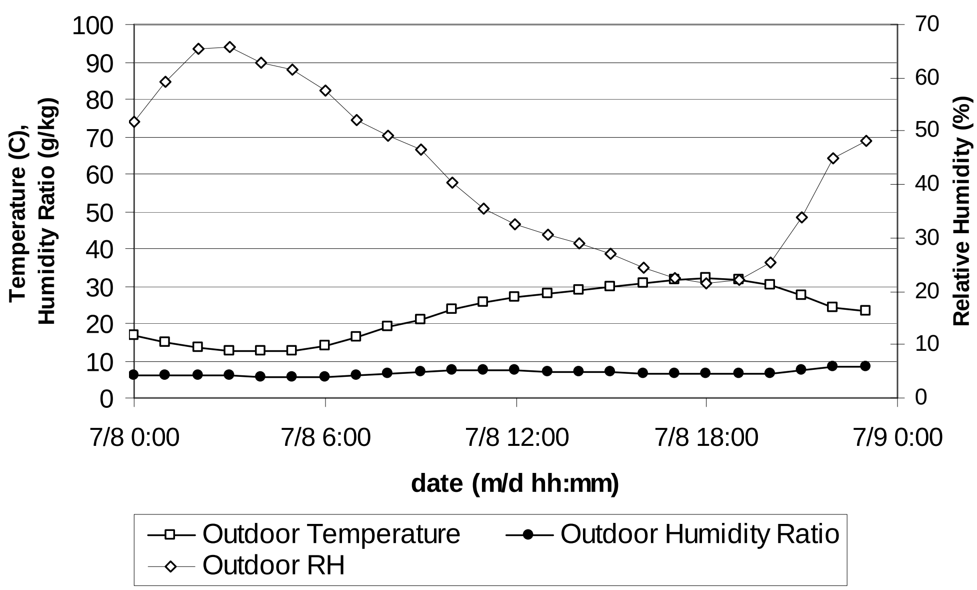


Figure 4.29. Hourly outdoor temperature, relative humidity and humidity ratio in Saskatoon on July 8th.

A comparison of PD_{tc} in Saskatoon is shown in Figure 4.30 for both December 16th and July 8th. The results for December 16th show very little difference (less than 1%) between the base case and the energy wheel case. The results for July 8th show a decrease of up to 4% when the energy wheel is used, compared to the base case.

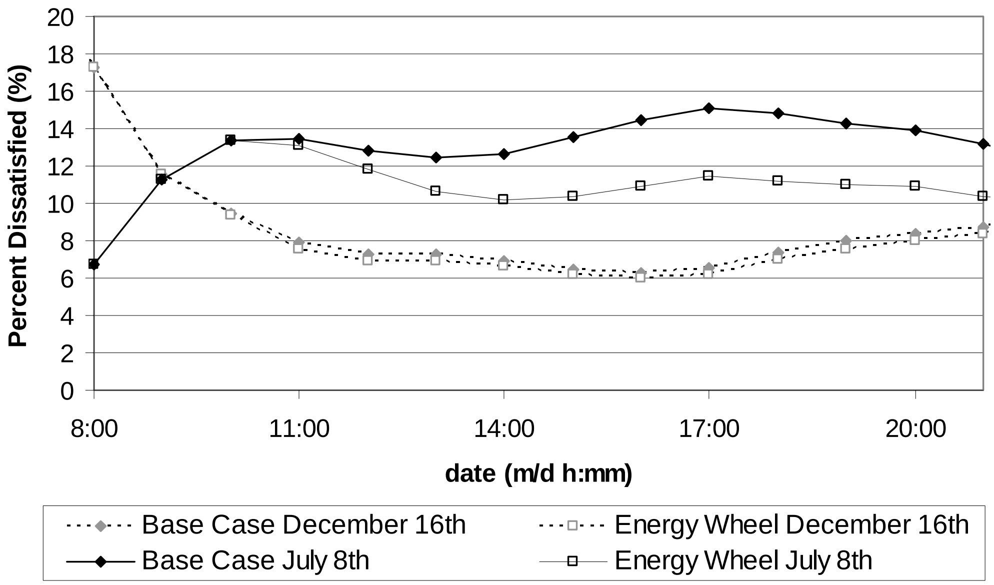


Figure 4.30. Hourly comparison of PD_{tc} in the office building in Saskatoon on December 16th and July 8th.

Figure 4.31 compares PD_{IAQ} for the base case and energy wheel case in Saskatoon on December 16th and July 8th. Once again, there is very little difference between the base case and the energy wheel case on December 16th and both simulations show a low value of PD_{IAQ}. The results on July 8th show a significant decrease in the percent dissatisfied for the energy wheel case, compared to the base case due to the large decrease in the RH levels in the space in the case with the energy wheel.

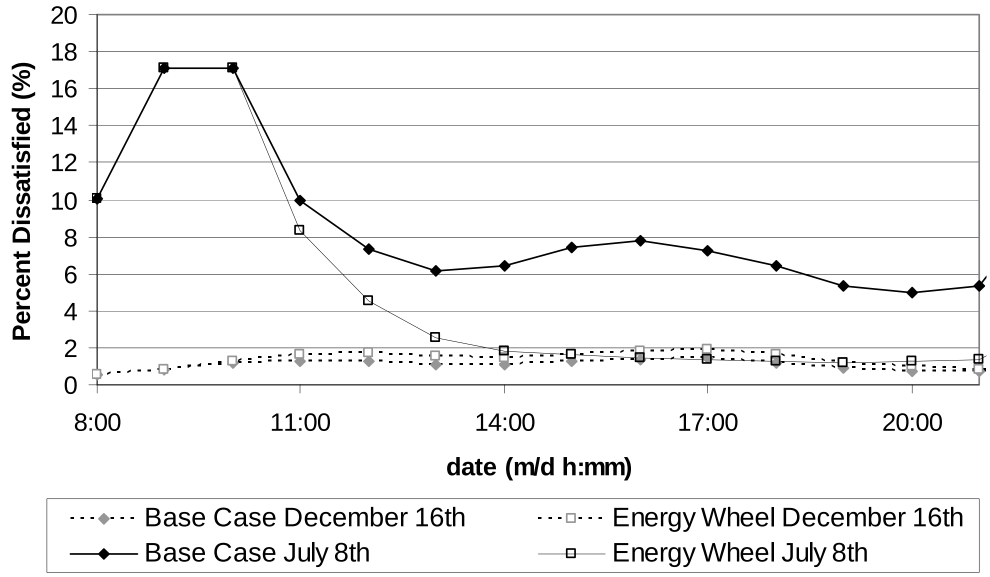


Figure 4.31. Hourly comparison of PD_{IAQ} in the office building in Saskatoon on December 16th and July 8th.

4.2.4 IAQ in the Office Building in Vancouver

The indoor temperature in Vancouver is quite similar to that of Saskatoon, with the temperature being maintained at 22°C during the winter and varying between 22°C and 24°C during the summer months as seen in Figure 4.32.

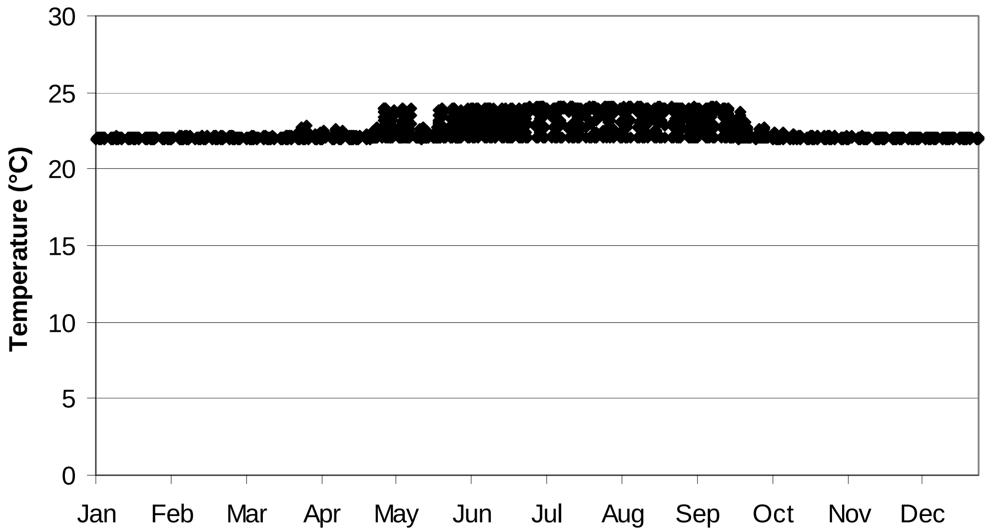


Figure 4.32. Indoor temperature in the office building in Vancouver during occupied hours.

The temperature difference between the base case and energy wheel case is shown for Vancouver in Figure 4.33. For a large portion of the hours (2000+) the temperature difference is 0°C. The maximum temperature difference is about 0.05°C and the minimum is about -0.04°C.

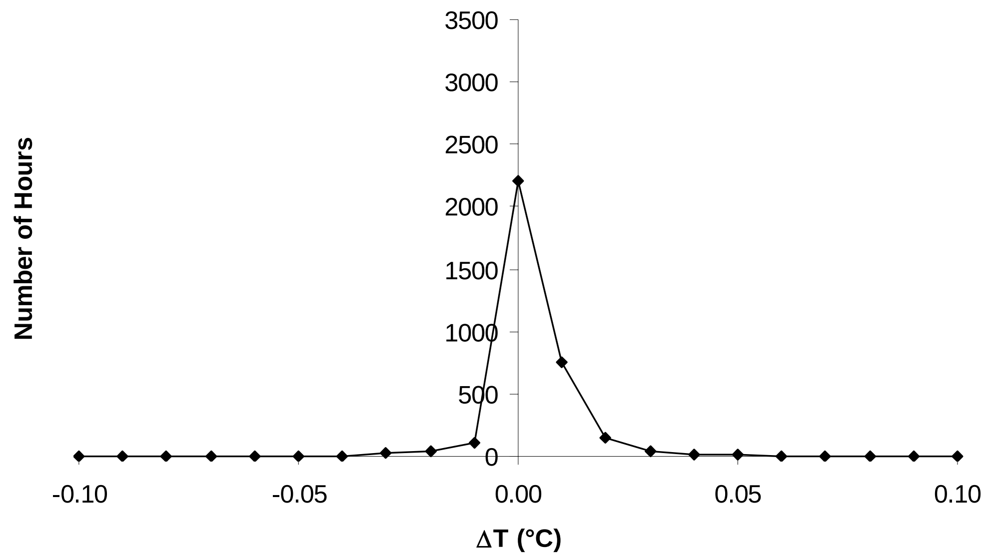


Figure 4.33. Frequency distribution of the difference between the indoor temperature calculated for the base case and energy wheel case in the office building in Vancouver.

The indoor relative humidity in Vancouver is shown in Figure 4.34. The relative humidity is low in the winter and high in the summer, similar to the results in Saskatoon.

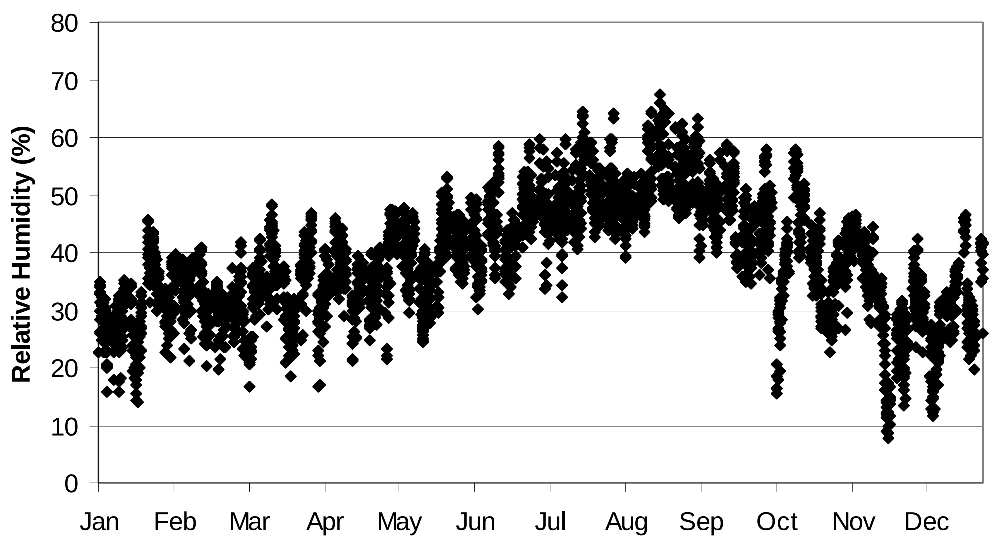


Figure 4.34. Indoor relative humidity in the office building in Vancouver during occupied hours.

Figure 4.35 shows the relative humidity differences between the base case and the energy wheel case. The differences here are very minimal. There are a large number of hours where the change in the relative humidity is around 0% RH . The maximum difference is about 0.1% RH and the minimum is around -0.1% RH. From these results it can be seen that in a mild climate such as Vancouver the addition of an energy wheel has little impact on PD_{tc} and PD_{IAQ} as can be seen in Figure 4.36.

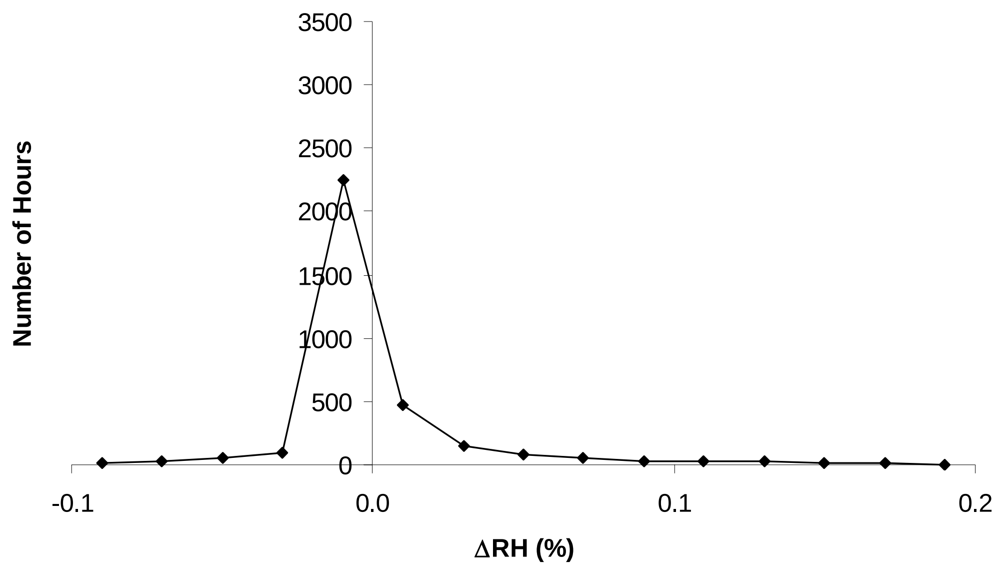


Figure 4.35. Frequency distribution of the difference between the indoor relative humidity calculated for the base case and energy wheel case in the office building in Vancouver.

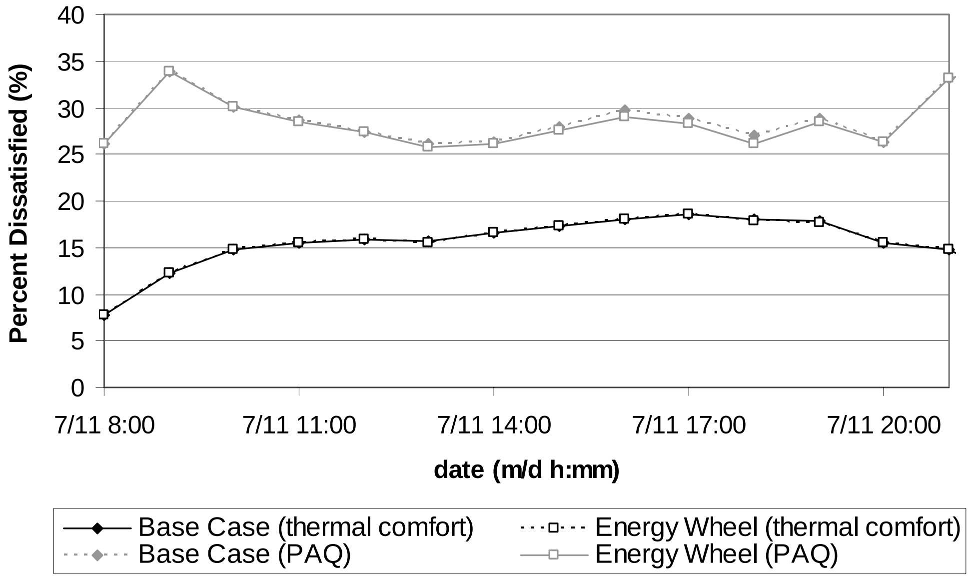


Figure 4.36. Hourly comparison of PD_{tc} and PD_{IAQ} in the office building in Vancouver on July 11th.

A summary of the relative humidity differences between the base case and energy wheel case for each of the four cities can be seen in Figure 4.37. The effect of adding an energy wheel on the relative humidity of the office space has the largest impact in Tampa and Phoenix. A small effect is noticeable in Saskatoon. Very little effect is noticeable in Vancouver.

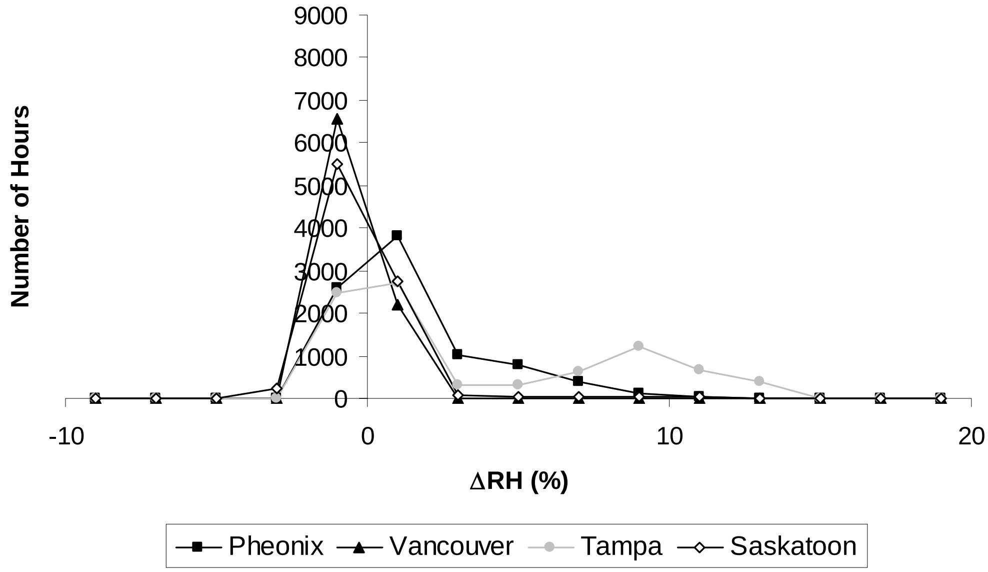


Figure 4.37. Summary of the relative humidity comparisons between the base case and energy wheel case in the office building for all four cities.

4.3 IAQ IN THE SCHOOL BUILDING

A summary of the difference in temperatures between the base case and the energy wheel case is presented in Figure 4.38 for the school. This is a summary over the occupied hours of the whole year. Again, a positive value indicates that the temperature was decreased with the addition of the energy wheel, over the base case and a negative value indicates that the temperature was increased. The frequency of ΔT is found over a range of values and then plotted at the average value of that range, so for Figure 4.38 a value plotted at -0.02°C is actually over the range of -0.04°C to 0°C. Also, it should be noted here that the school has only 1566 occupied hours during the year.

The ΔT values shown in Figure 4.38 are very small. The range of temperature differences is approximately -0.2°C to 0.2°C with the majority of values (up to 83% in Vancouver) falling around the 0°C mark.

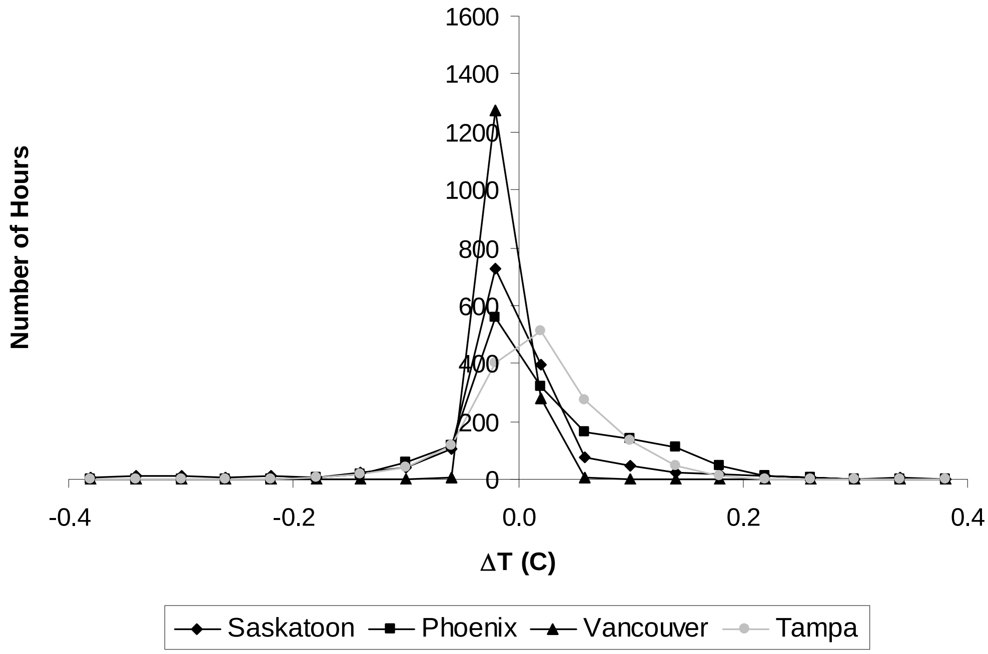


Figure 4.38. Summary of the indoor temperature comparisons between the base case and energy wheel case in the school for all four cities.

Figure 4.39 shows a summary of the difference in indoor relative humidity between the base case and energy wheel case for the school. The energy wheel has a large impact on the indoor RH in Tampa, showing a peak around 15% RH and a maximum difference of about 30% RH. In Saskatoon the energy wheel has a small impact on the indoor RH values, showing a decrease of about 6% RH during the summer months and an increase of about 9% RH in the winter months. The results for Phoenix show an increase in the indoor RH, up to 15% RH. This occurs mostly during the winter months when the indoor RH during the base case gets as low as 20% RH. This increase is actually good because the desired range of indoor RH is 30-70% RH. In these cases where the base case RH is around 20% RH the addition of the energy wheel increases the indoor RH to about 35% RH, which is in the acceptable range. Vancouver shows little difference in the indoor RH, with about 71% of the values falling around the 0% RH mark.

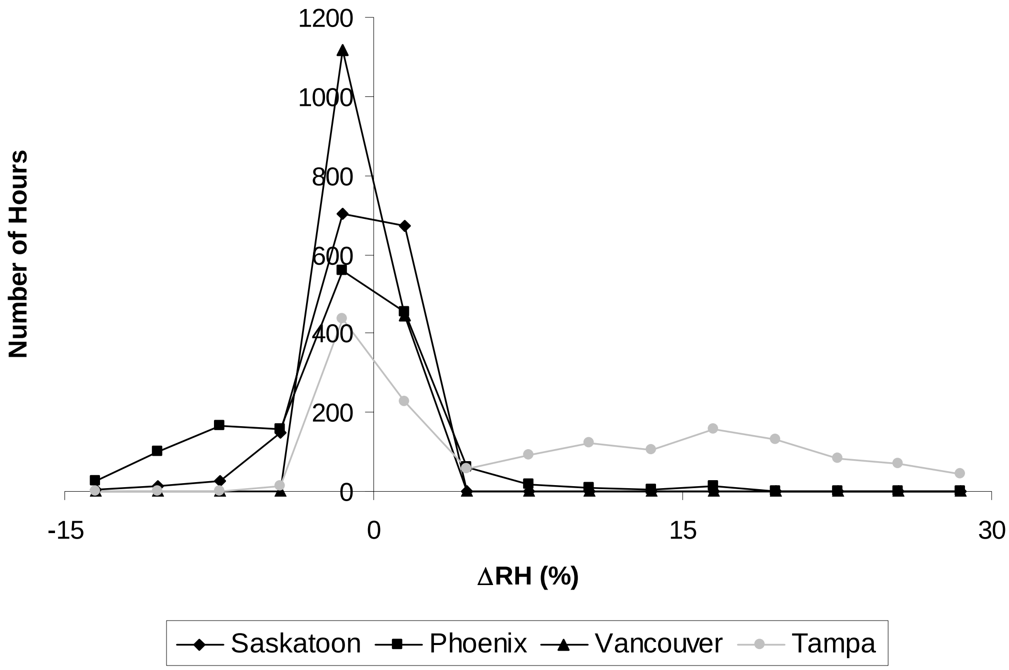


Figure 4.39. Summary of the indoor relative humidity comparisons between the base case and energy wheel case in the school for all four cities.

The percent dissatisfied with thermal comfort and PAQ are shown in Figures 4.40 and 4.41, respectively. All four cities show similar results for ΔPD_{tc}, with values ranging from approximately -1% to 1.5%. Although there is a slight improvement in PD_{tc}, it is not a large enough difference to be considered significant. Figure 4.41 shows a significant decrease in PD_{IAQ} with the addition of an energy wheel in Tampa with a peak difference around 15% and a maximum difference around 30%. Saskatoon, Vancouver and Phoenix all show a difference of around 0% for the majority of occupied hours.

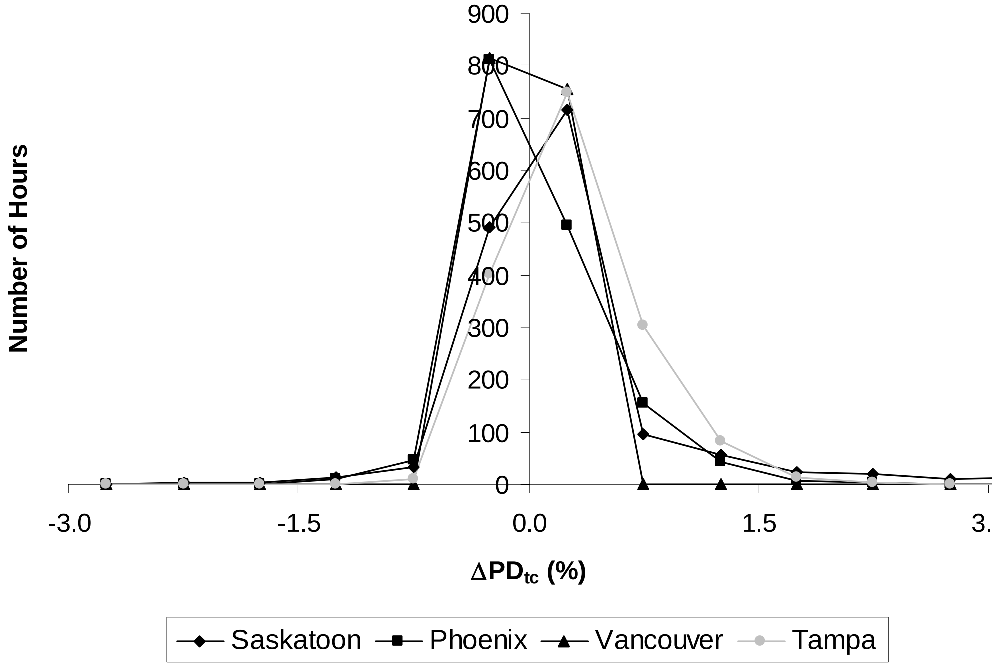


Figure 4.40. Summary of PD_{tc} comparisons between the base case and energy wheel case in the school for all four cities.

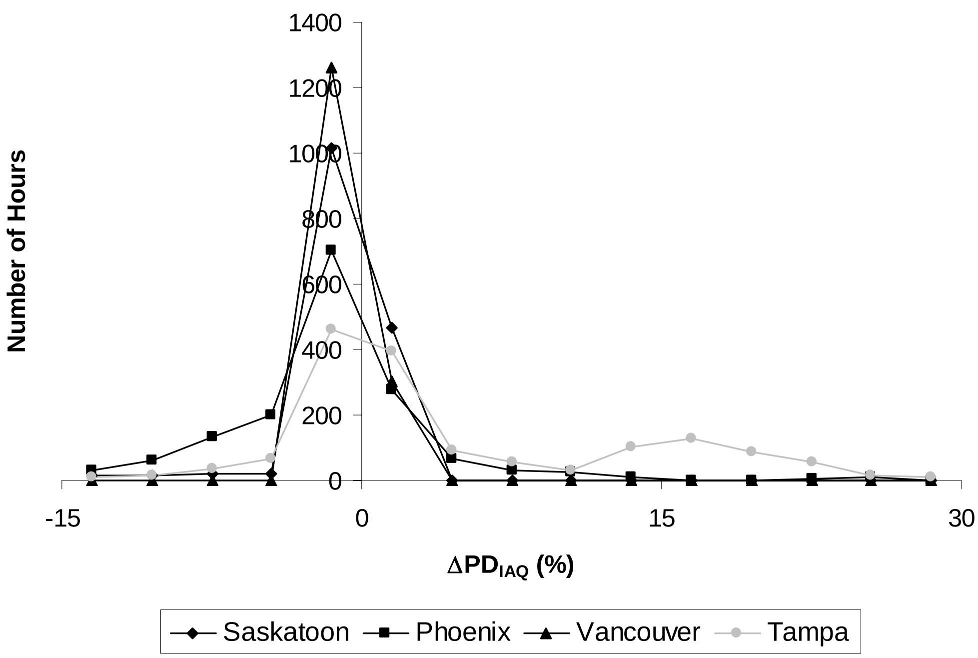


Figure 4.41. Summary of PD_{IAQ} comparisons between the base case and energy wheel case in the school for all four cities.

CHAPTER 5

ENERGY CONSUMPTION AND COST ANALYSIS

This chapter will present the effect of an energy wheel on heating energy (natural gas) and cooling energy (electricity) consumption as well as HVAC equipment capacities. The energy consumption and equipment capacity results of the base case will also be compared to Dhital (1994). Finally, a cost and payback analysis is done to determine if the addition of an energy wheel into a building in each climate is economically feasible.

5.1 EFFECT OF A VARIABLE EFFECTIVENESS ENERGY WHEEL -ENERGY

As with the IAQ results in Chapter 4, the constant effectiveness energy wheel simulation is first compared to the variable effectiveness energy wheel simulation for the office building. Figure 5.1 shows the natural gas and electricity consumption for each of the simulations in Saskatoon. The variable effectiveness case shows a small decrease in the amount of natural gas consumed by the office building as compared to the constant variable case. There is a small difference in the amount of electricity consumed by the two cases, but it is so small that it cannot be seen on the graph.

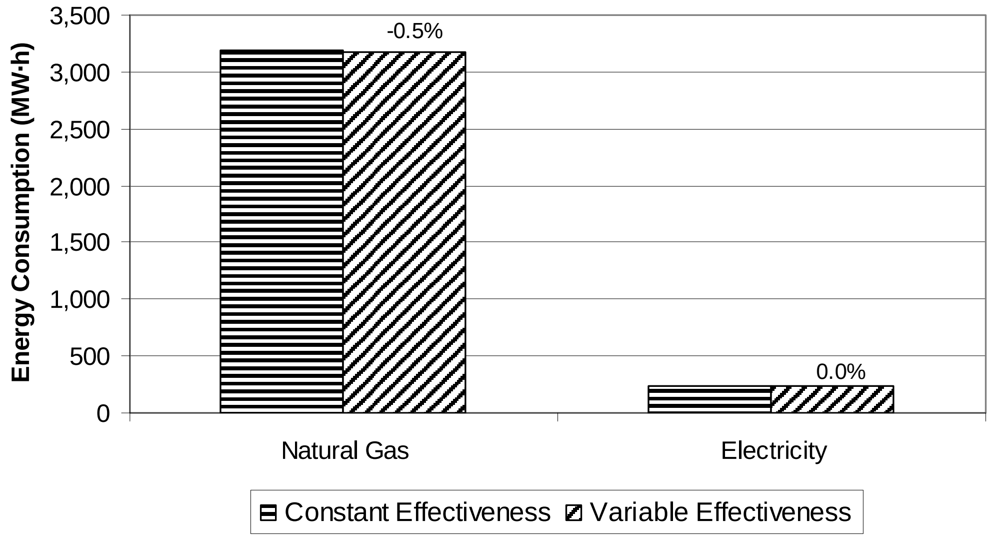


Figure 5.1. Comparison of energy consumption for the constant and variable effectiveness cases in Saskatoon.

Figure 5.2 shows the equipment sizes for the constant effectiveness and variable effectiveness case. The variable effectiveness case shows a small increase in the size of the cooling coil required, as compared to the constant variable case. There is a small change in the size of the boiler between the two cases, but again, it is too small to be seen on the graph. From these results it can be determined that there is no significant difference between the constant and variable effectiveness cases and so the base case results will be compared to the constant effectiveness case only.

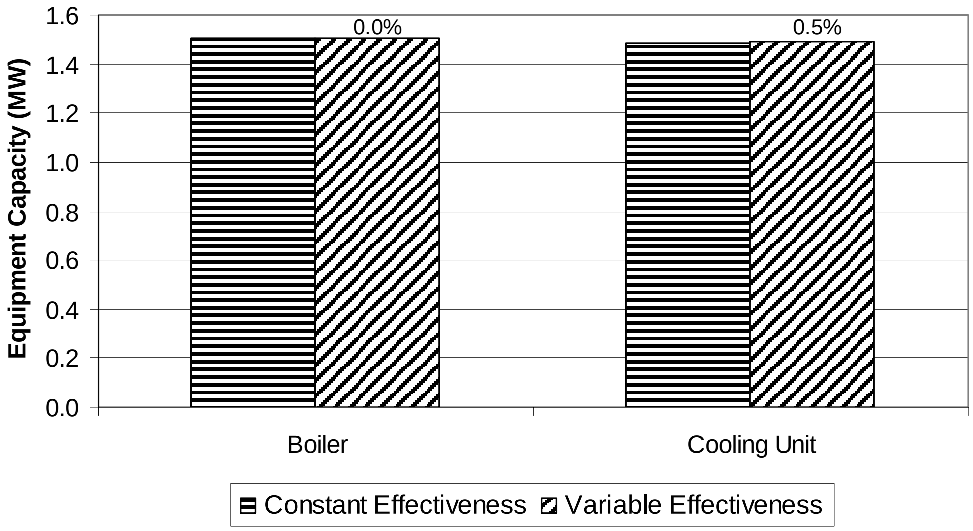


Figure 5.2. Comparison of equipment size for the constant and variable effectiveness cases in Saskatoon.

5.2 ENERGY ANALYSIS FOR THE OFFICE BUILDING

5.2.1 Energy Consumption for the Office Building - Base Case

While it has been shown that energy wheels can help improve the indoor air quality by changing the relative humidity in the space, energy wheels are traditionally installed to reduce energy consumption and the cost of heating and cooling buildings. In order to determine the feasibility of energy wheels in each of the four cities investigated here, the amount of energy consumed and cost of heating and cooling in each city must be determined.

The heating system uses natural gas for heating and the vapor-compression cooling system uses electricity for cooling. A detailed description of the HVAC system is given in Section 2.3. Electricity is also required for lights, and equipment in the space as well as the fans in the HVAC system. Figure 5.3 shows the total annual energy consumption for each of the cities. The natural gas consumption varies from 384 MWh (56MJ/m^2) in Tampa to 3,540 MWh (520MJ/m^2) in Saskatoon. This is expected because of the large difference in the climates. The electricity consumption varies from 3,666 MWh (536MJ/m^2) in Vancouver to 5,785 MWh (843MJ/m^2) in Tampa. Since the lighting and equipment requirements will be the same for each city and the fan requirements are very similar for each city, the large difference is due to the cooling load.

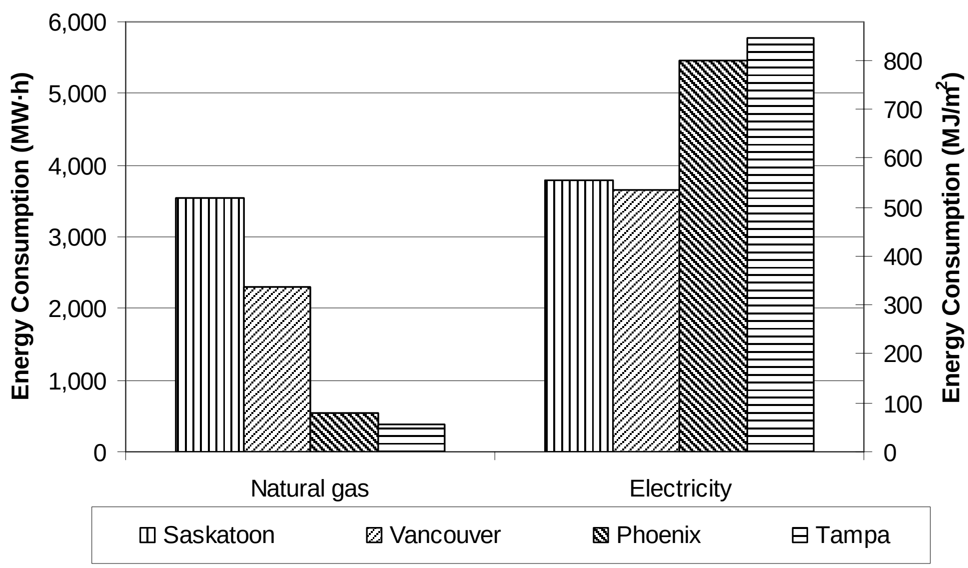


Figure 5.3. Total annual energy consumption for the base case office building in all cities.

Figure 5.4 shows the fraction of the annual energy consumption due to the major components (heating, cooling, lighting, fans and equipment) for the base case in each city. The cooling load ranges from 2% in Vancouver to 36% in Tampa. The heating load ranges from 6% in Tampa to 48% in Saskatoon. In Saskatoon, the total energy consumed for conditioning of the air (heating and cooling) is 52% of the total energy consumed by the building on a yearly basis. Conditioning the air accounts for 41% of the energy consumed in Vancouver, 42% in Phoenix and 43% in Tampa. Since the fans are also included in the HVAC system it is more accurate to give the total percentage of energy consumed in each city including the fans. The total energy consumed for conditioning and ventilating the building becomes 69% in Saskatoon, 62% in Vancouver, 62% in Phoenix and 63% in Tampa.

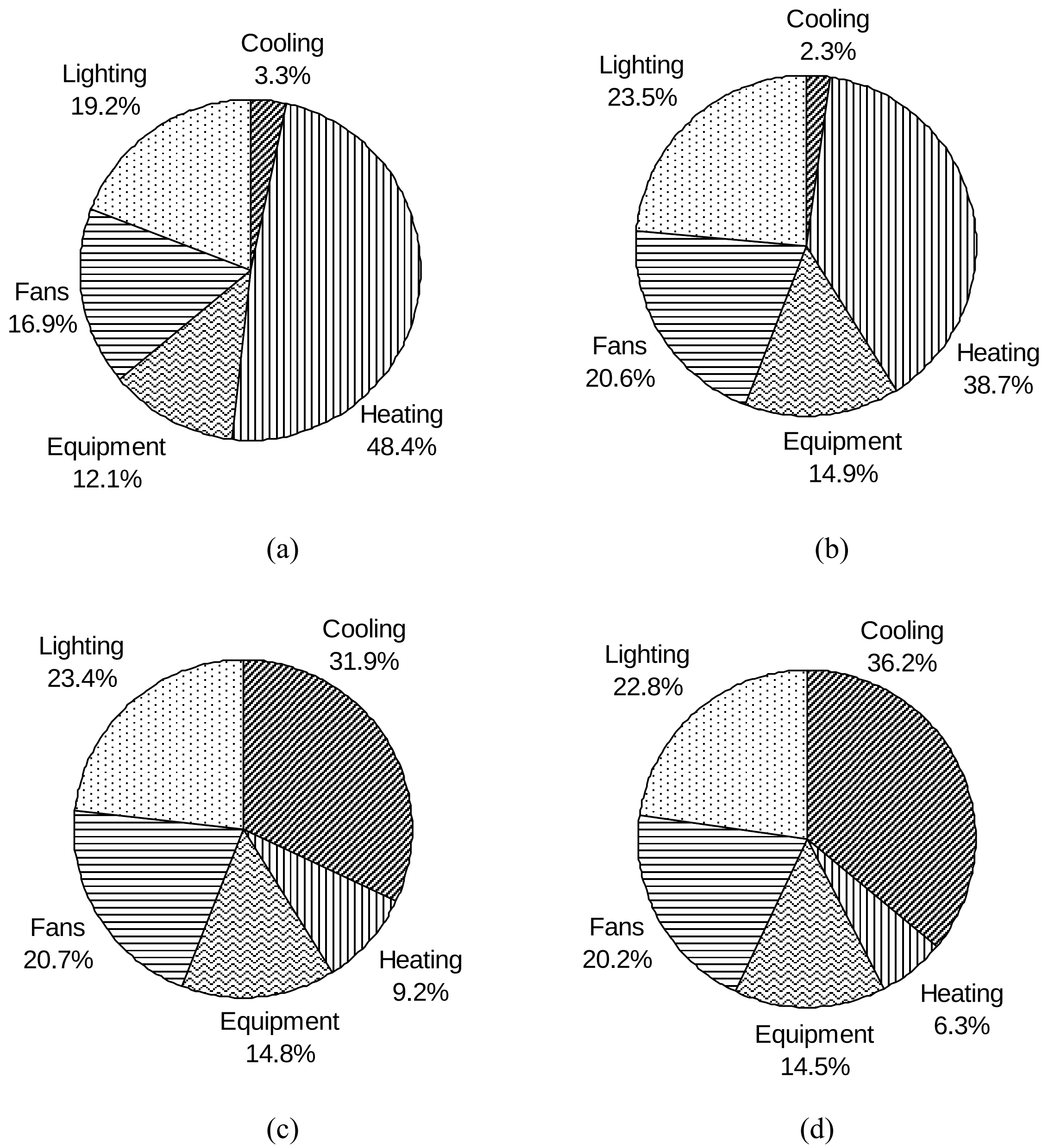


Figure 5.4. Distribution of annual energy consumption for the base case office building in (a) Saskatoon, (b) Vancouver, (c) Phoenix and (d) Tampa.

Since the office building is based on the work of Dhital (1994) it is useful to compare the current TRNSYS results of the base case simulation with those of Dhital (1994). To allow a more direct comparison some of the assumptions presented in Chapter 2 are modified to more closely match the input used by Dhital (1994). The only changes made to permit this comparison are the number of people in the lobby is increased from 20 to 210 and the number of computers in the office is increased from 1537 to 2554. The results of these simulations can be seen in Figure 5.5. It should be noted that the results of Dhital (1994) are for Edmonton and the simulations performed in this thesis are for Saskatoon. Edmonton and Saskatoon have similar but slightly different climates. Considering the difference in climates and slight differences in the building that still remain the results are quite similar, which adds confidence to the energy consumption and cost data that will be presented in the following sections.

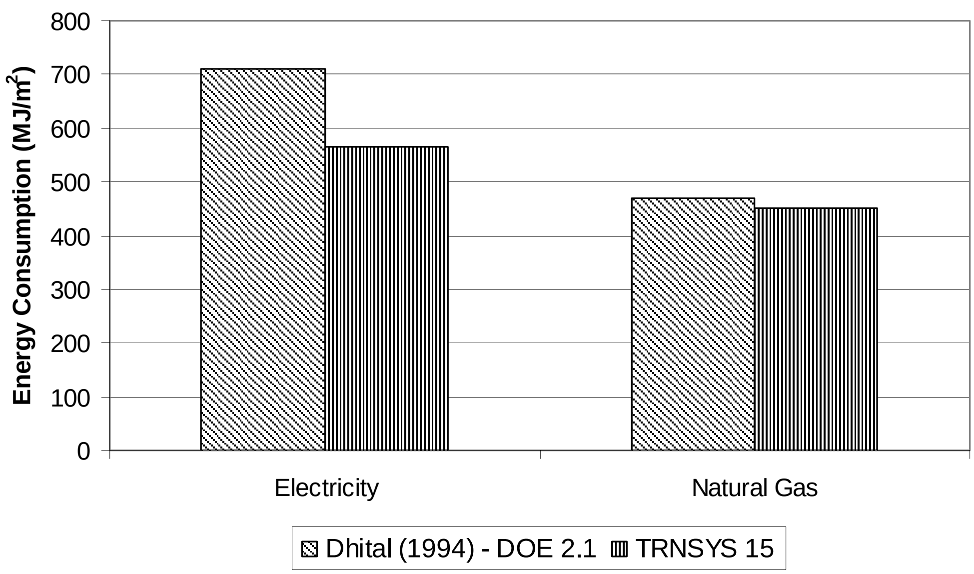


Figure 5.5. Comparison between the annual energy consumption calculated using TRNSYS in this study with the DOE 2.1 results of Dhital (1994).

5.2.2 Energy Consumption of the Office Building

The total amount of natural gas consumed in each city is shown in Figure 5.6 for the base case and the energy wheel case. The most significant change is in Saskatoon where the natural gas consumption decreases by 10.2% with the addition of the energy wheel, as compared to the base case. The difference in the natural gas consumption between the two cases for Vancouver is very minimal and does not show on the chart. Phoenix and Tampa both experienced very slight increases (0.1%) with the addition of the energy wheel. No change is expected in these two cities as they are warmer climates and do not have much of a heating season.

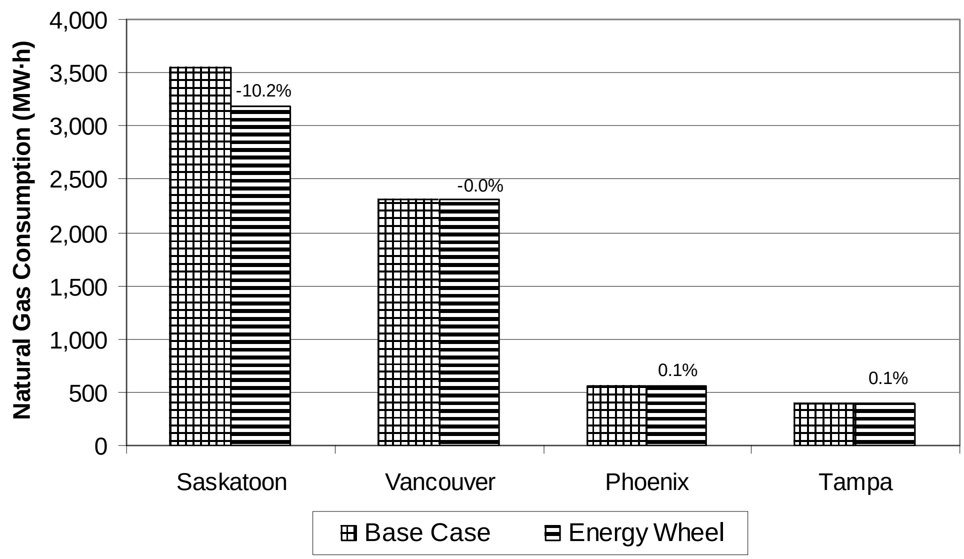


Figure 5.6. Comparison of the total annual natural gas consumption in the office building.

Figure 5.7 shows that the total overall electrical energy consumption, including cooling, fans, lights and equipment. The overall electrical energy consumption is reduced by 4.4% in Phoenix and 4.8% in Tampa. This reduction results because the decrease in the electrical energy required for cooling is larger than the increase in electricity consumption of the fans. The 200 Pa (0.8 inH_2O) drop across the energy wheel results in a typical increase in fan energy consumption of 4 MW·h per year. Figure 5.7 shows that the net change in electrical energy consumption is very small in Saskatoon and Vancouver. Figure 5.8 shows the annual electrical energy required for cooling alone. There is a large reduction in the amount of cooling required in Phoenix and Tampa, 13.2% and 13.1%, respectively and a slight decrease in Saskatoon (1.9%). The reduction in Saskatoon is smaller because it has a colder climate and does not require as much cooling. There is a very small increase in the cooling consumption in Vancouver, possibly due to the larger amount of heat added by the fans when the energy wheel is in the HVAC system.

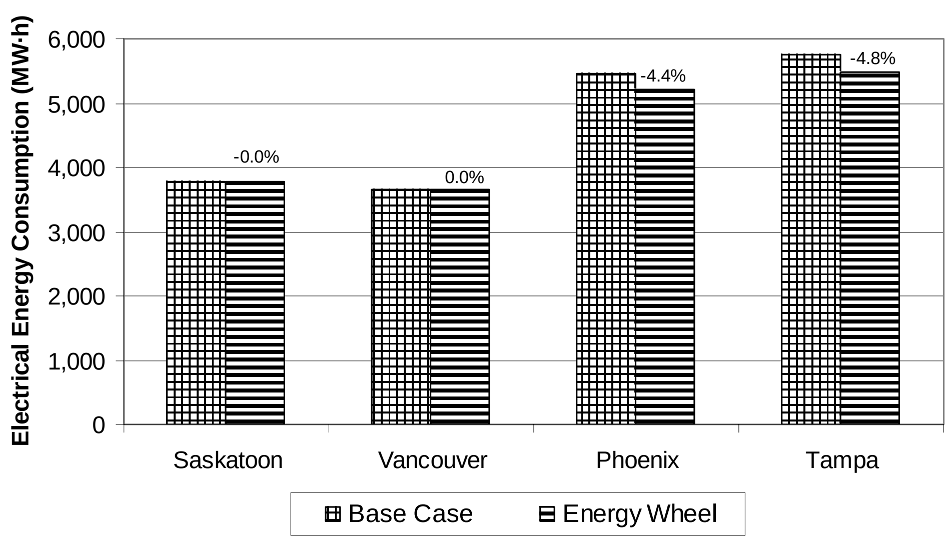


Figure 5.7. Comparison of the total annual electrical energy consumption in the office building.

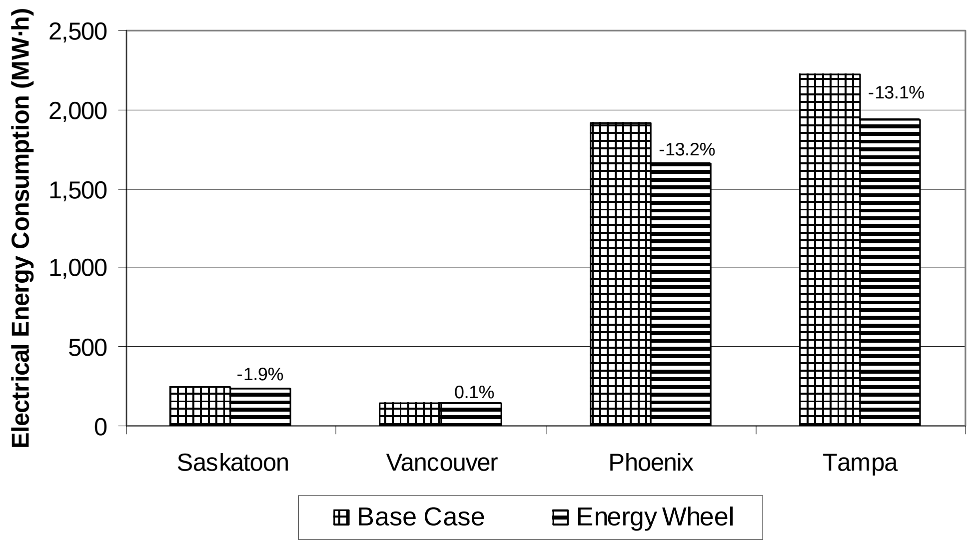


Figure 5.8. Comparison of the annual electrical energy consumed for cooling in the office building.

The annual energy consumption broken down into components for the energy wheel case can be seen in Figure 5.9. The cooling load ranges from 2.3% in Vancouver to 32.9% in Tampa. The cooling load in Vancouver is the same as the base case, but the cooling load in Tampa has decreased from the base case. The heating load ranges from 6.6% in Tampa to 45.7% in Saskatoon. The heating load percentages in Tampa and Phoenix have increased because the heating load has stayed the same, but the total energy consumed has decreased. The heating load percentage in Saskatoon has decreased from the base case.

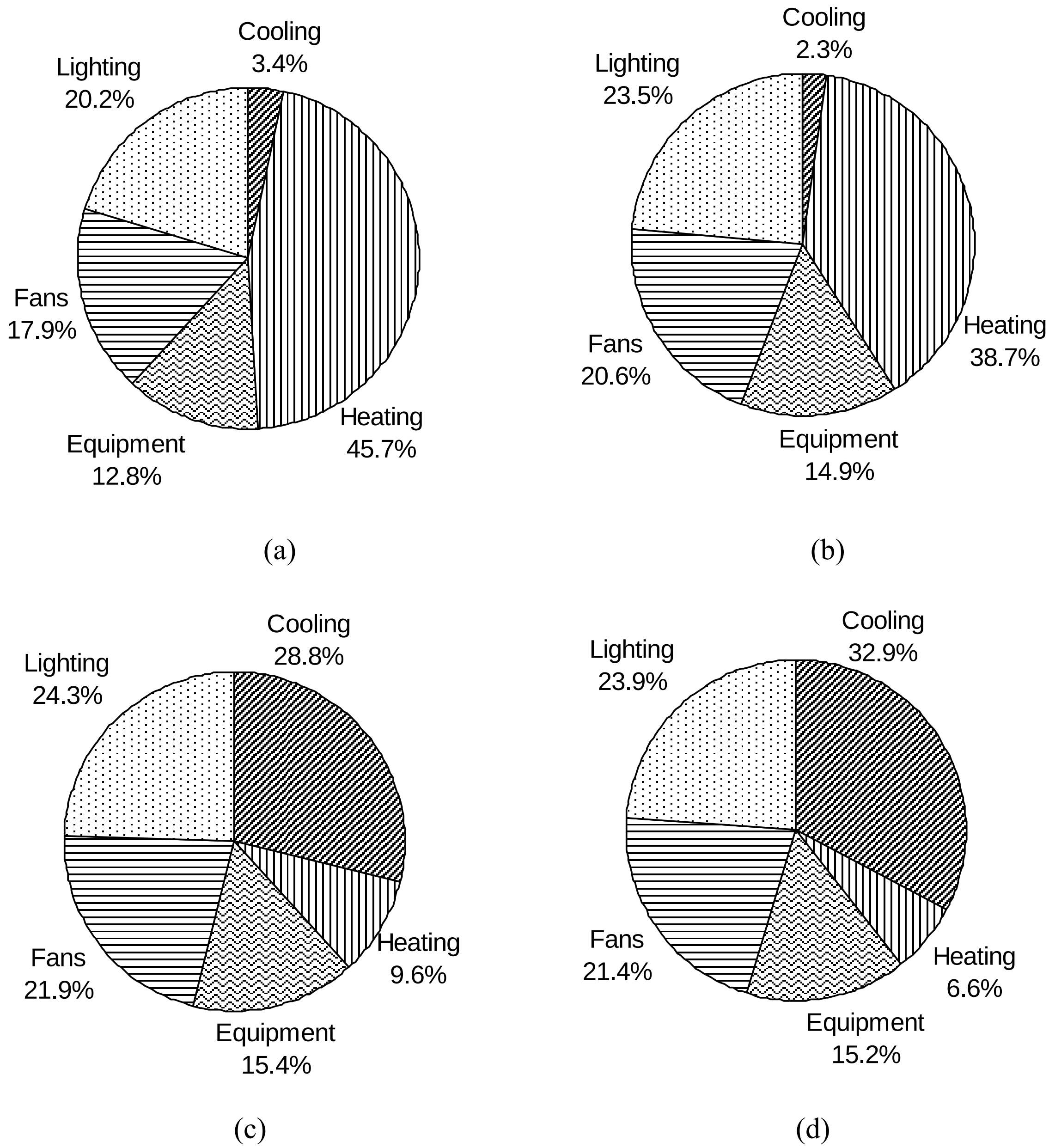


Figure 5.9. Distribution of annual energy consumption for the energy wheel case office building in (a) Saskatoon, (b) Vancouver, (c) Phoenix and (d) Tampa.

The energy consumed for heating and cooling the air is 49% of the total energy consumed in Saskatoon, 41% in Vancouver, 38% in Phoenix and 40% in Tampa. The fraction of total energy that is consumed by the HVAC system to condition and ventilate the building (including fans) is 67% in Saskatoon, 62% in Vancouver, 60% in Phoenix and 61% in Tampa. These values are summarized in Table 5.1 for the base case and energy wheel case. The total amount of energy consumed for conditioning of the air has decreased in all the cities except for Vancouver. This means that putting an energy wheel into the HVAC system of a building in Saskatoon, Phoenix and Tampa will help reduce the operating costs associated with conditioning and ventilating the building.

Table 5.1. Comparison of the energy consumed for conditioning of the air in the office building as a percentage of the total energy consumption in each city.

	Heating and Cooling		Heating, Cooling and Fans	
	Base Case	Energy Wheel	Base Case	Energy Wheel
Saskatoon	52%	49%	69%	67%
Vancouver	41%	41%	62%	62%
Phoenix	42%	38%	62%	60%
Tampa	43%	40%	63%	61%

5.2.3 Equipment Capacities for the Office Building

The boiler and cooling unit capacities that are required is determined by the peak energy transfer between the equipment and the air. The required boiler capacity for both the base case and the energy wheel case can be seen in Figure 5.10. The energy wheel reduces the required boiler capacity by 25.5% in Saskatoon, indicating that an energy wheel may be cost effective in Saskatoon. The energy wheel has very little impact on the required boiler capacity in the other cities due to their warm climates.

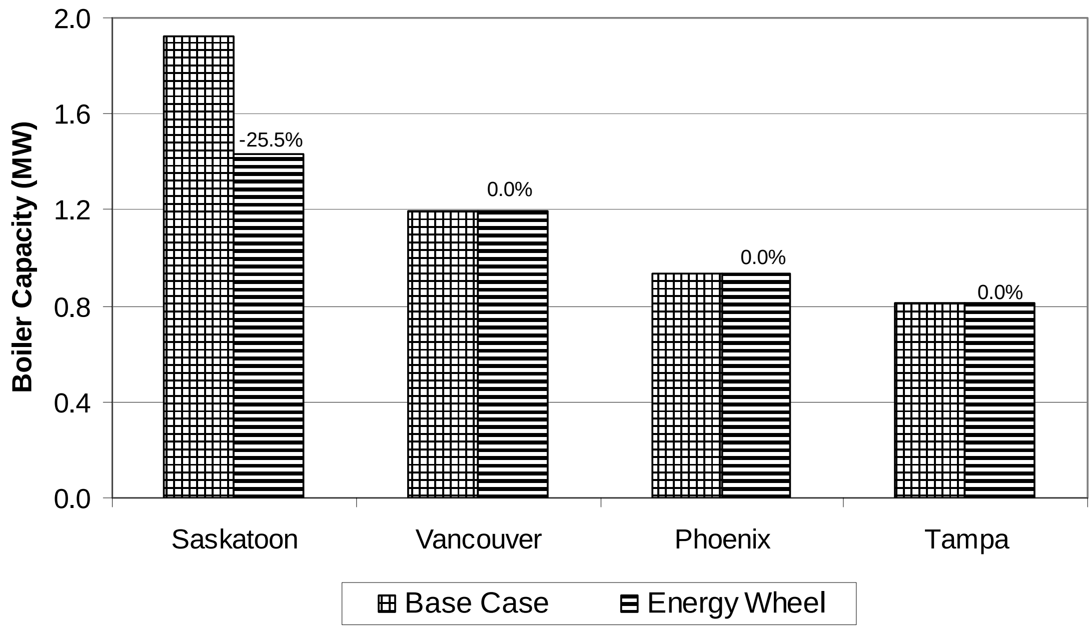


Figure 5.10. Comparison of boiler capacity required to heat the office building with and without an energy wheel.

The capacity of the cooling unit required to cool the building and ventilation air for the base and energy wheel cases are found in Figure 5.11. With the energy wheel there is a small reduction in the capacity of the cooling unit in Saskatoon (4%) because Saskatoon does not have a very large cooling load, but there is a large reduction in both Phoenix (18%) and Tampa (17%). From this it appears that an energy wheel will provide cost benefits in Phoenix and Tampa. There is a very small increase of 0.1% in the capacity of the cooling unit in Vancouver when the energy wheel is added. This slight increase in cooling capacity could be due to the extra heat added to the supply and exhaust streams by the larger fans needed to overcome the pressure drop across the energy wheel. This again shows that an energy wheel would not be beneficial in Vancouver.

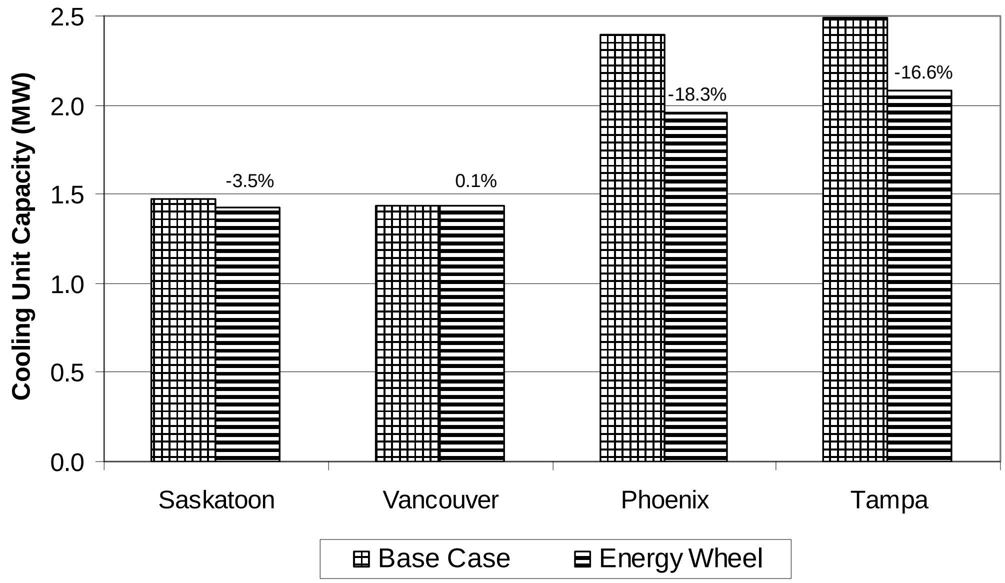


Figure 5.11. Comparison of cooling unit capacity required to cool the office building with and without an energy wheel.

Again, the results of these simulations are compared to the results of Dhital (1994) in Figure 5.12 for the heating and cooling equipment capacities. The results show good agreement for both the boiler and the cooling unit.

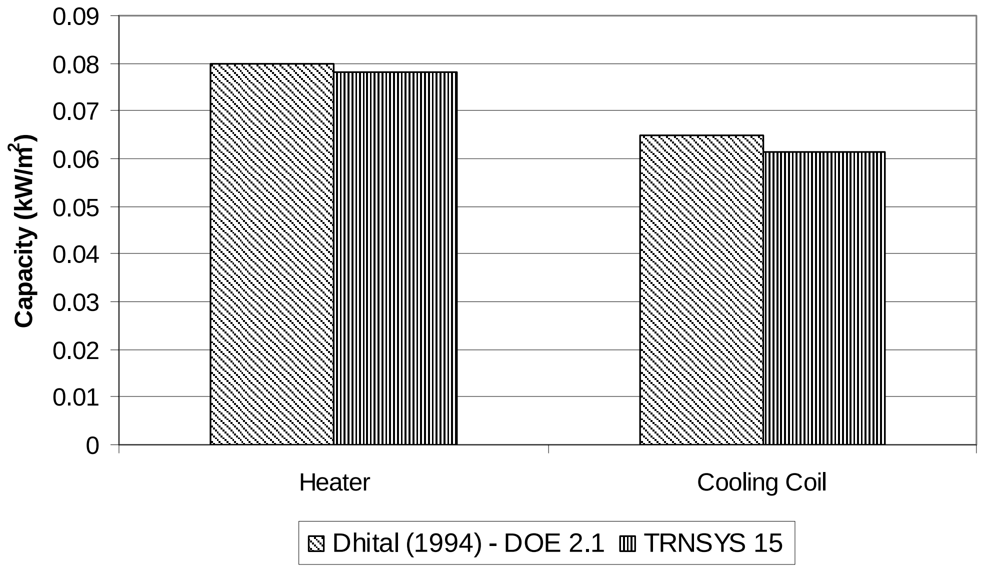


Figure 5.12. Comparison between the heating and cooling equipment capacity calculated using TRNSYS in this study with the DOE2.1 results of Dhital (1994).

5.2.4 Cost Analysis for the Office Building

A cost analysis is performed on the office building based on the work of Asiedu et al. (2005). There are two main parameters that need to be calculated to determine if the HVAC system with an energy wheel is economically viable. These parameters are the life-cycle cost and payback period of the system. The life-cycle cost is calculated by

$$C_{tot} = C_{ex} + C_{eq} + C_{fan} + C_{aux}, \tag{5.1}$$

where C_{tot} = life-cycle cost
C_{ex} = cost of energy wheel
C_{eq} = cost of auxiliary heating and cooling equipment
C_{fan} = cost of fan power
C_{aux} = cost of auxiliary heating and cooling power.

The cost of the auxiliary heating and cooling equipment is assumed to be \$200/kW (Asiedu et al., 2005) and the cost of the energy wheel is assumed to be \$6/(L/s). The cost of the auxiliary heating and cooling power is

$$C_{aux} = P_{wef}\left(e_c q_{tot,c} + e_h q_{tot,h}\right), \tag{5.2}$$

where P_{wef} is the present worth escalation factor, e_c and e_h are the cost of electricity and natural gas, respectively and $q_{tot,c}$ and $q_{tot,h}$ are the total electrical energy requirements for cooling and the total natural gas requirements for heating the building. The present worth escalation factor is taken from Asiedu et al. (2005) and given a value of 8.61 (based on a system life of 10 years, an interest rate of 6% and an escalation rate of 3%). The cost of the fan power is

$$C_{fan} = P_{wef} e_c q_{tot,fan}, \tag{5.3}$$

where $q_{tot,fan}$ is the total electrical energy required by the fans. Table 5.2 shows the cost of electricity (City of Saskatoon, 2006) and natural gas (SaskPower, 2006) used in equations (5.2) and (5.3).

Table 5.2. Cost of electricity and natural gas.

	Electricity (\$/kW·h)	Natural Gas (\$/kW·h)
first 16,750kWh	0.0883	0.0803
balance	0.0565	0.0514

Table 5.3 shows the life cycle cost (LCC) of the base case and energy wheel systems over a 10 year life cycle. LCC savings are realized in Saskatoon (5.8%), Phoenix (4.4%) and Tampa (4.9%) with an energy wheel, but not in Vancouver. In Vancouver, adding an energy wheel to the HVAC system will increase the LCC by over \$95,000. It

turns out this is the capital cost of the energy wheel, which again shows that no energy savings are achieved in Vancouver.

Table 5.3. Life-cycle cost of the base case and energy wheel case HVAC systems.

		Total Life Cycle Cost
Saskatoon	Base Case	$2,986,761
	Energy Wheel	$2,812,824
	Savings	**$173,938 (5.8%)**
Vancouver	Base Case	$2,228,367
	Energy Wheel	$2,323,601
	Savings	**-$95,234 (-4.3%)**
Phoenix	Base Case	$2,463,913
	Energy Wheel	$2,354,770
	Savings	**$109,142 (4.4%)**
Tampa	Base Case	$2,532,072
	Energy Wheel	$2,409,375
	Savings	**$122,697 (4.9%)**

The payback period of the system is amount of time, in years, in which savings must be realized to overcome the capital cost of the modifications to the HVAC system. This is calculated by

$$PB = \frac{C_{ex} - C_{aux,N} + C_{aux,ex}}{(C_{rec} - \Delta C_{fan})/P_{wef}}, \tag{5.4}$$

where C_{rec} is the savings due to heating and cooling recovered in the energy wheel, ΔC_{fan} is the increase in the cost of the fan due to the increased pressure drop created by the energy wheel, $C_{aux,N}$ is the cost of auxiliary heating and cooling equipment if there is no energy recovery (base case) and $C_{aux,ex}$ is the cost of auxiliary heating and cooling equipment with an energy wheel. The payback period is calculated for each of the four cities and is shown in Table 5.4. Phoenix and Tampa show very good payback periods, less than one year. Vancouver has a much larger payback period because there are very little savings with the energy wheel. As the life of the HVAC system is assumed to be 10 years, the installation of an energy wheel in a building in Vancouver will never pay for itself. Saskatoon shows a negative payback period because the capital cost of the HVAC equipment is actually smaller when the energy wheel is included due to the reduction in heating and cooling equipment size. Therefore, the payback in Saskatoon is immediate and any future benefits (energy savings and improved thermal comfort and PAQ) result from essentially no investment.

Table 5.4. Payback period for the energy wheel in each of the four cities.

	Payback Period (years)
Saskatoon	-0.69
Vancouver	56,491
Phoenix	0.54
Tampa	0.79

5.3 ENERGY ANALYSIS FOR THE SCHOOL BUILDING

5.3.1 Energy Consumption for the School Building - Base Case

The annual energy consumption for the school building is shown in Figure 5.13 for the base case. The natural gas consumption ranges from 580 MW·h in Tampa to 3370 MW·h in Saskatoon. The electrical energy consumption ranges from 1297 MW·h in Vancouver to 1951 MW·h in Tampa. The heating loads in the school are slightly higher than in the office building for each particular city. The cooling loads in the school however, are about one-third of the cooling loads in the office building for each city.

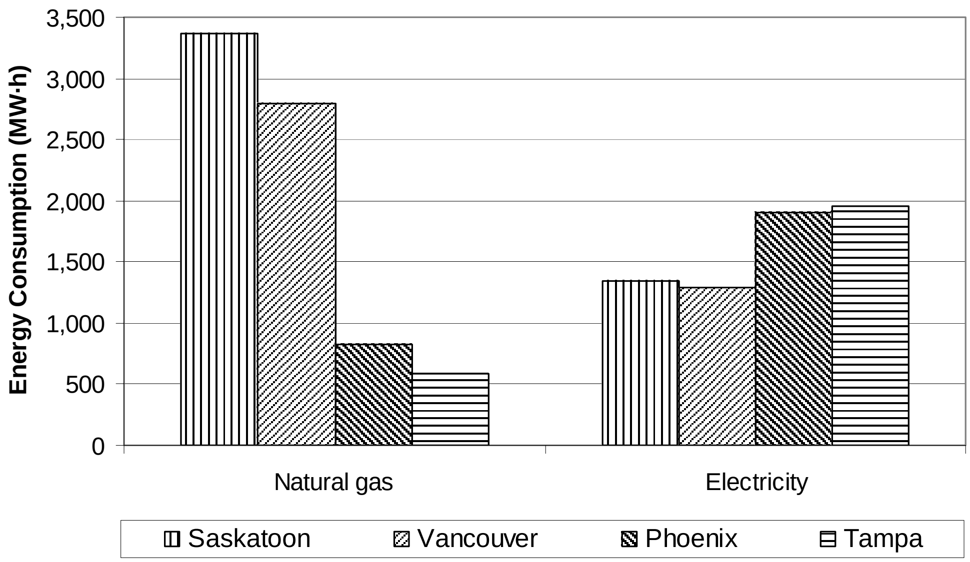


Figure 5.13. Annual energy consumption for the base case school building in all cities.

The annual energy consumption distribution for the school building is shown in Figure 5.14 for each city. In the school building, the HVAC system requires a much larger percent of the total energy consumption than in the office building. For the base case school building, the fraction of total energy that is required for heating ranges from 23% in Tampa to 71% in Saskatoon. The fraction of total energy that is required for cooling ranges from 0.1% in Vancouver to 23% in Tampa. The energy consumption due to heating and cooling together makes up 72%, 68%, 51% and 46% of the overall energy consumption in Saskatoon, Vancouver, Phoenix and Tampa respectively. With the fans included, the fraction of energy consumed for conditioning and ventilating the school is 96% in Saskatoon, 96% in Vancouver, 94% in Phoenix and 93% in Tampa.

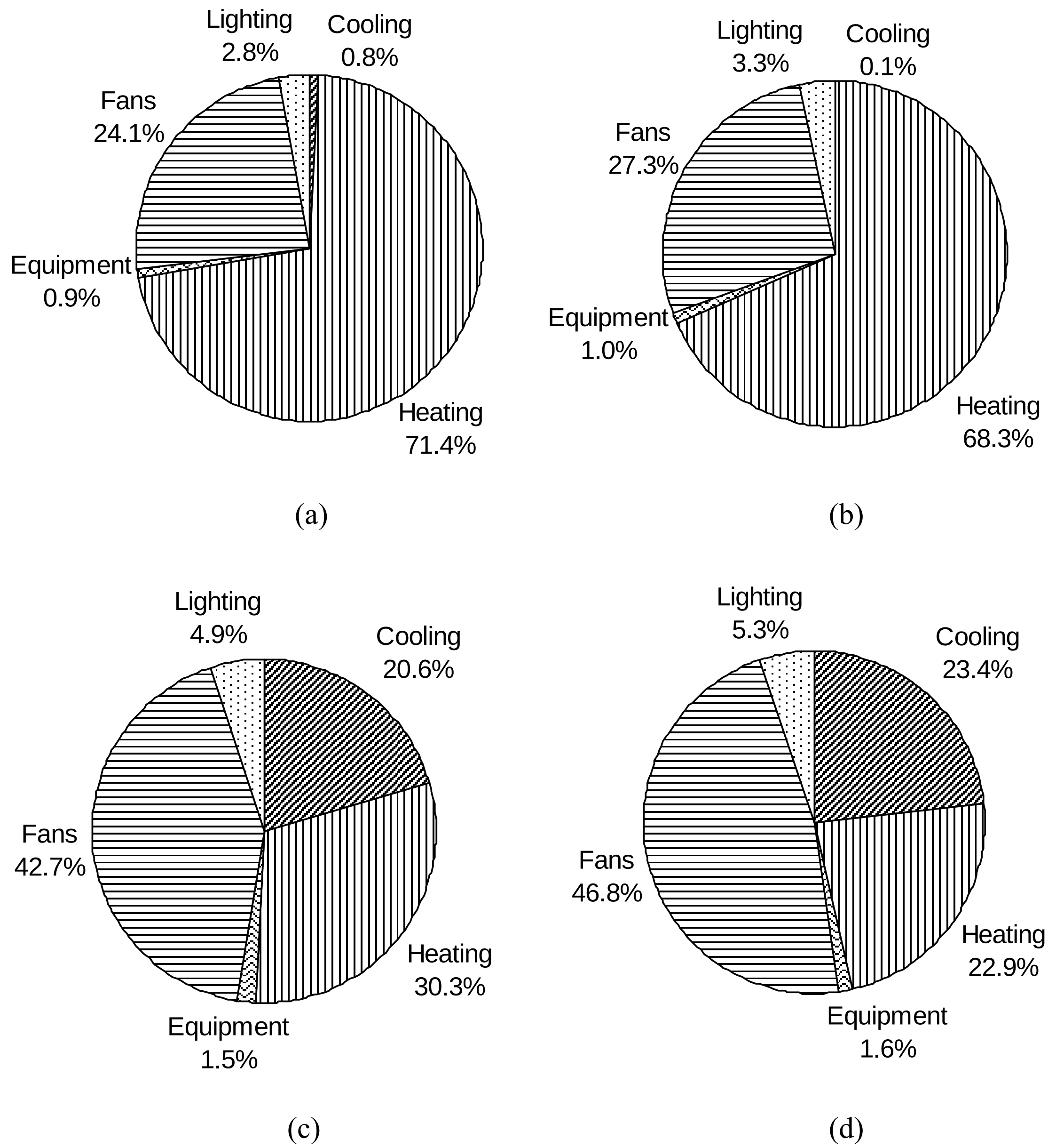


Figure 5.14. Distribution of annual energy consumption for the base case school building in (a) Saskatoon, (b) Vancouver, (c) Phoenix and (d) Tampa.

5.3.2 Energy Consumption of the School Building

The annual natural gas consumption for the energy wheel is compared to the base case in Figure 5.15 for the school building in each city. There is an 8% reduction in the amount of natural gas consumed by the school building in Saskatoon when the energy wheel is added to the system. Tampa and Phoenix show a very small increase in the amount of natural gas consumed in the energy wheel case, as compared to the base case. There is virtually no change in the natural gas consumption in Vancouver.

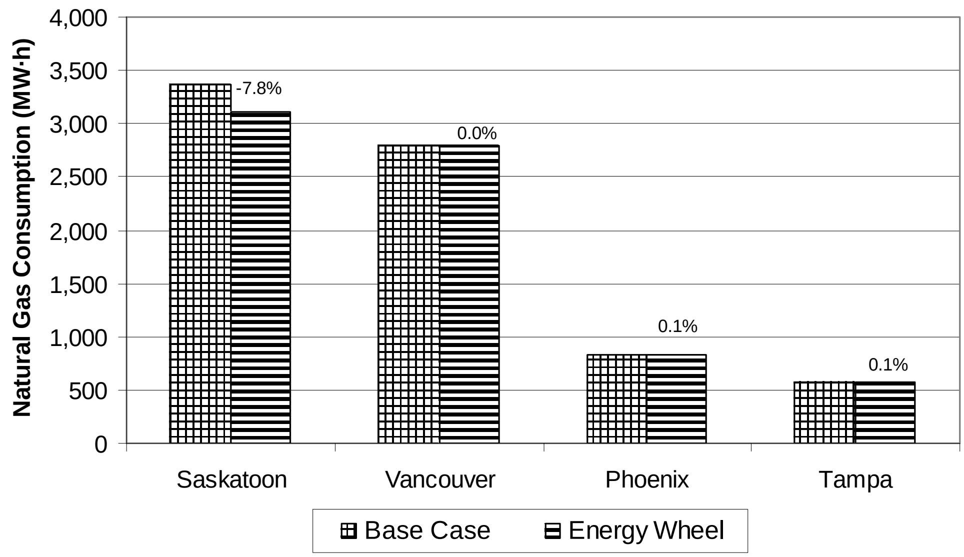


Figure 5.15. Comparison of the annual natural gas consumption in the school building.

Figure 5.16 shows the annual electrical energy consumption in the school building for the base case and the energy wheel case. There is a very minimal reduction in the electrical energy consumed in the energy wheel case for Saskatoon and Vancouver, compared to the base case, while the reduction is significant in Phoenix (10%) and Tampa (11%).

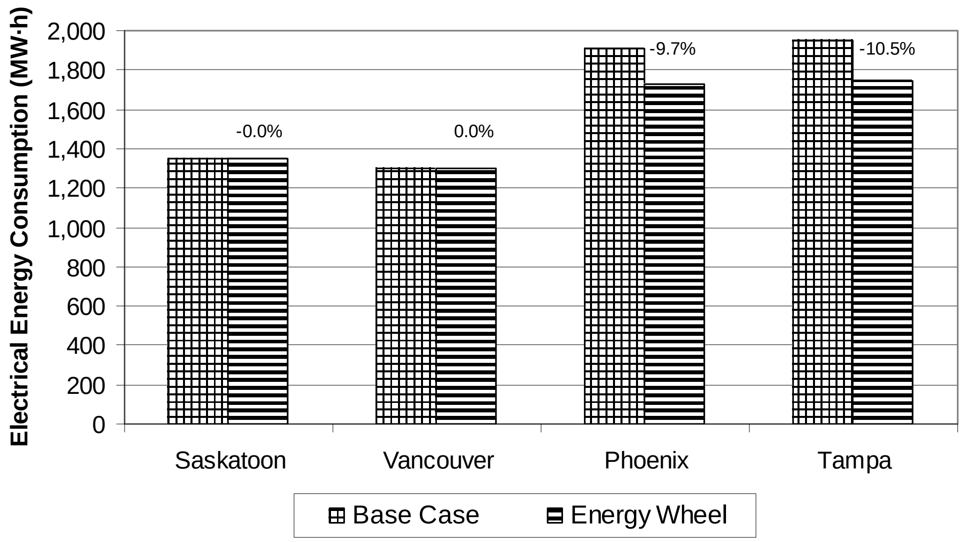


Figure 5.16. Comparison of the annual electrical energy consumption in the school building.

The annual electrical energy consumed for cooling in the base case and energy wheel case is shown in Figure 5.17. The largest reduction in electrical energy consumed for cooling occurs in Phoenix, with the consumption being reduced 33% with the addition of an energy wheel. There is a reduction of 32% in Tampa, 14% in Saskatoon and 1% in Vancouver.

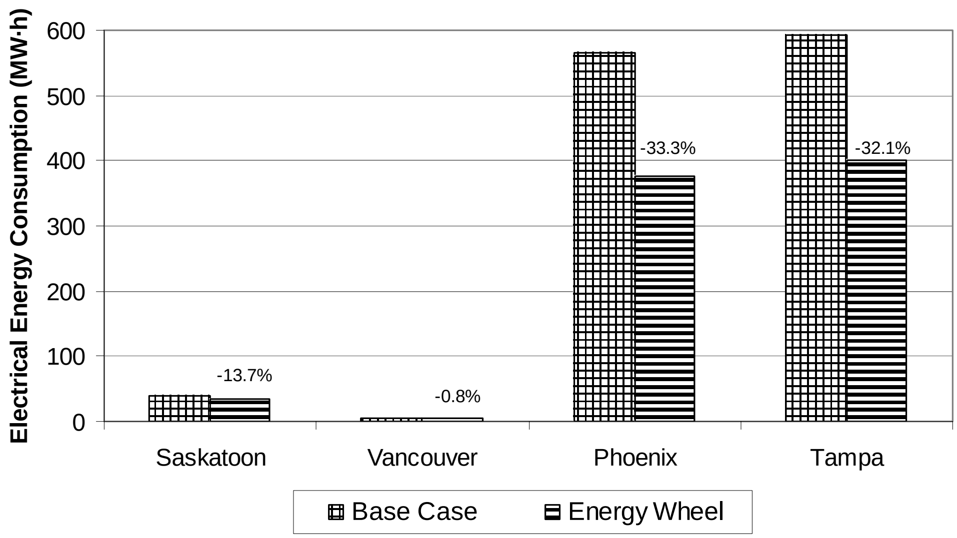
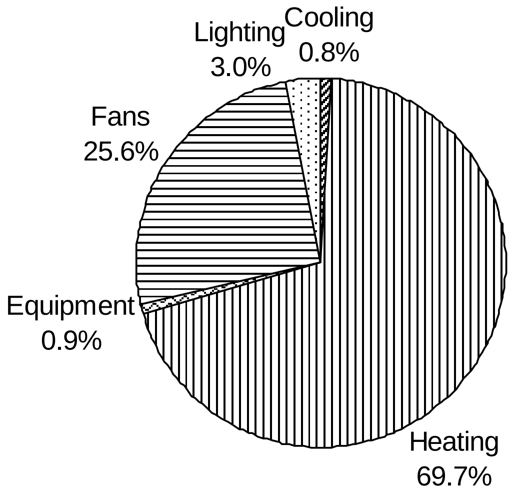
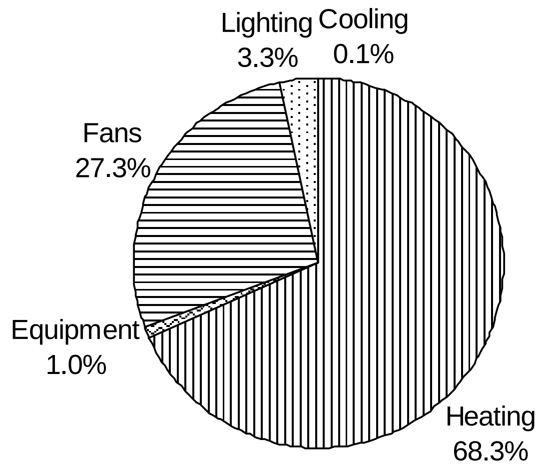


Figure 5.17. Comparison of the annual electrical energy consumed for cooling in the school building.

The annual energy distribution for the energy wheel case is shown for each city in Figure 5.18. The percentage of the total energy consumption that is required by the HVAC system to ventilate and condition the school (fans, heating and cooling) is 96% in Saskatoon, 96% in Vancouver, 93% in Phoenix and 93% in Tampa. In Saskatoon, Tampa and Phoenix the energy wheel has a positive impact on the energy consumption in the school building. This is summarized in Table 5.5. The energy wheel has no impact on the energy consumption of the school building in Vancouver.



(a)



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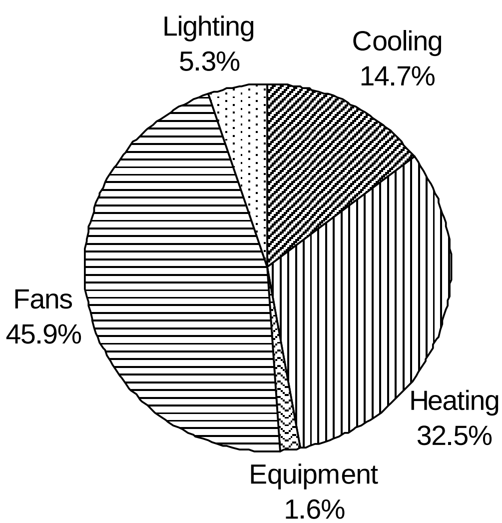

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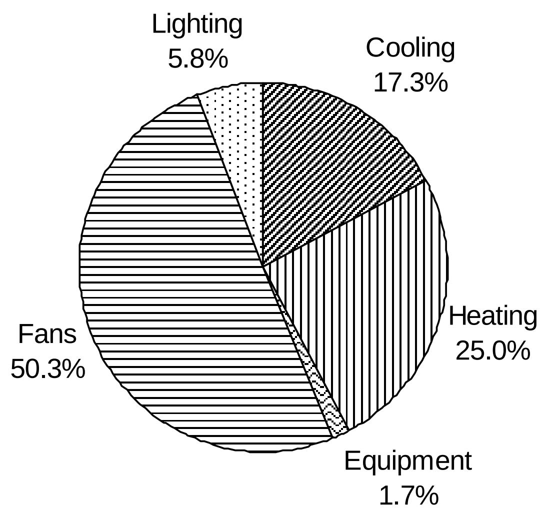

(d)

Figure 5.18. Distribution of annual energy consumption for the energy wheel case school building in (a) Saskatoon, (b) Vancouver, (c) Phoenix and (d) Tampa.

Table 5.5. Comparison of the energy consumed for conditioning of the air in the school building as a percentage of the total energy consumed for each city.

	Heating and Cooling		Heating, Cooling and Fans	
	Base Case	Energy Wheel	Base Case	Energy Wheel
Saskatoon	72%	71%	96%	96%
Vancouver	68%	68%	96%	96%
Phoenix	51%	47%	94%	93%
Tampa	46%	42%	93%	93%

5.3.3 Equipment Capacities for the School Building

In Saskatoon, there is a large reduction in the size of the heating equipment (28%), when the energy wheel is added to the HVAC system as seen in Figure 5.19. The other three cities do not show any significant change in the size of heating system required to condition the building with and without an energy wheel.

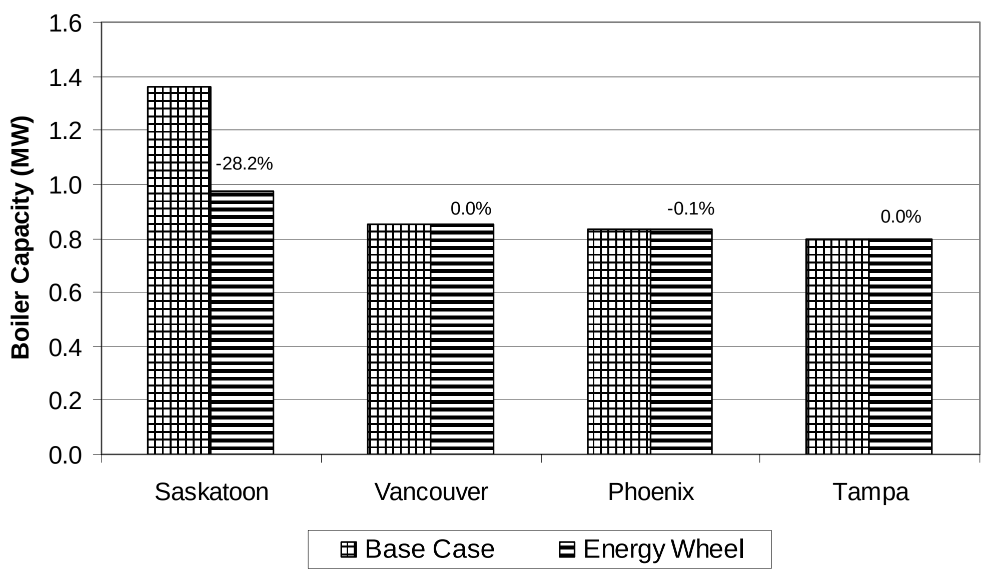

Figure 5.19. Comparison of boiler capacity for the school building.

Figure 5.20 shows the required capacities of the cooling unit for each city. Again, there are large reductions in Saskatoon (16.4%), Phoenix (33%) and Tampa (27%) when the energy wheel is incorporated into the system. There is no significant impact of the energy wheel on the size of cooling unit required in Vancouver.

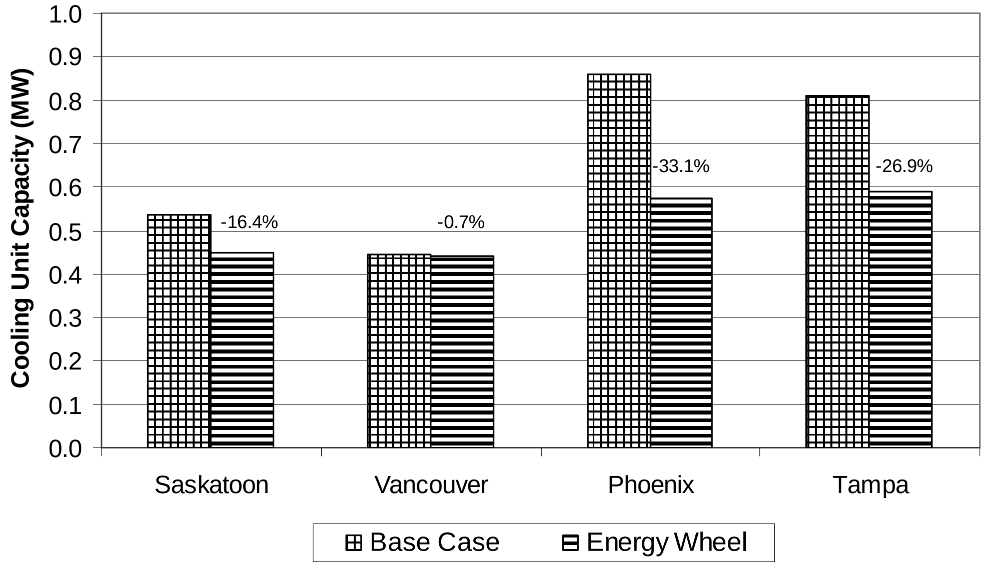

Figure 5.20. Comparison of cooling unit capacity for the school building.

5.3.4 Cost Analysis for the School Building

The life-cycle costs of the HVAC systems in the base case and energy wheel case are shown in Table 5.6. Once again savings are realized in Saskatoon, Phoenix and Tampa, but the savings are not as large as with the office building. In Vancouver, the life cycle cost is greater by the cost of the energy wheel ($95,000).

Table 5.6. Life cycle costs of the school building in the base case and energy wheel case.

		Total Life Cycle Cost
Saskatoon	Base Case	$2,455,971
	Energy Wheel	$2,340,105
	Savings	**$115,866**
Vancouver	Base Case	$2,055,430
	Energy Wheel	$2,149,921
	Savings	**-$94,491**
Phoenix	Base Case	$1,564,218
	Energy Wheel	$1,512,473
	Savings	**$51,745**
Tampa	Base Case	$1,455,194
	Energy Wheel	$1,406,914
	Savings	**$48,280**

Table 5.7 shows the payback period for the energy wheel in the school building in each of the four cities. Saskatoon has a payback period of a few days, which means the cost of the HVAC system is essentially the same with and without the energy wheel. The payback periods in Phoenix (3.64 years) and Tampa (4.44 years) are also very attractive. Vancouver does not have a payback period because the cost of the heating and cooling energy actually increases with the addition of the energy wheel and with no savings being realized, the system will never be paid back.

Table 5.7. Payback period of the school building in each of the four cities.

	Payback Period (years)
Saskatoon	0.05
Vancouver	------
Phoenix	3.64
Tampa	4.44

CHAPTER 6

SENSITIVITY STUDY

This chapter investigates the sensitivity of the results to several different factors: the amount of recirculation air, the energy wheel effectiveness, the building construction and the outdoor ventilation rates. The sensitivity study will be compared with the office building.

6.1 SENSITIVITY OF THE FRACTION OF RECIRCULATION AIR

The fraction of recirculation air used in the previous simulations is 0.8 (80%) under normal operating conditions. To determine how sensitive the simulations are to this value, a study is conducted where the recirculation air fraction is reduced to 0.7 (70%) in the office building with an energy wheel. All other variables are kept the same including the outdoor ventilation rate. Figure 6.1 shows the difference in temperatures between the 70% case and the 80% case in all climates. The range of temperature differences is between -1.5°C and 1.5°C.

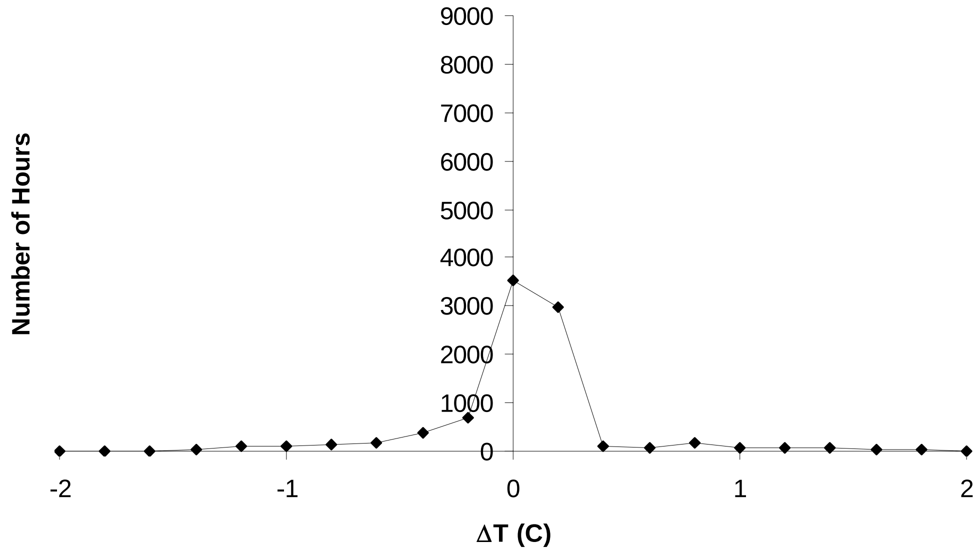

Figure 6.1. Temperature comparison for the fraction of recirculation air study in Saskatoon.

The relative humidity differences from the 80% case to the 70% case are shown in Figure 6.2. The difference in relative humidity ranges from -5% RH to 10% RH. This is a significant difference.

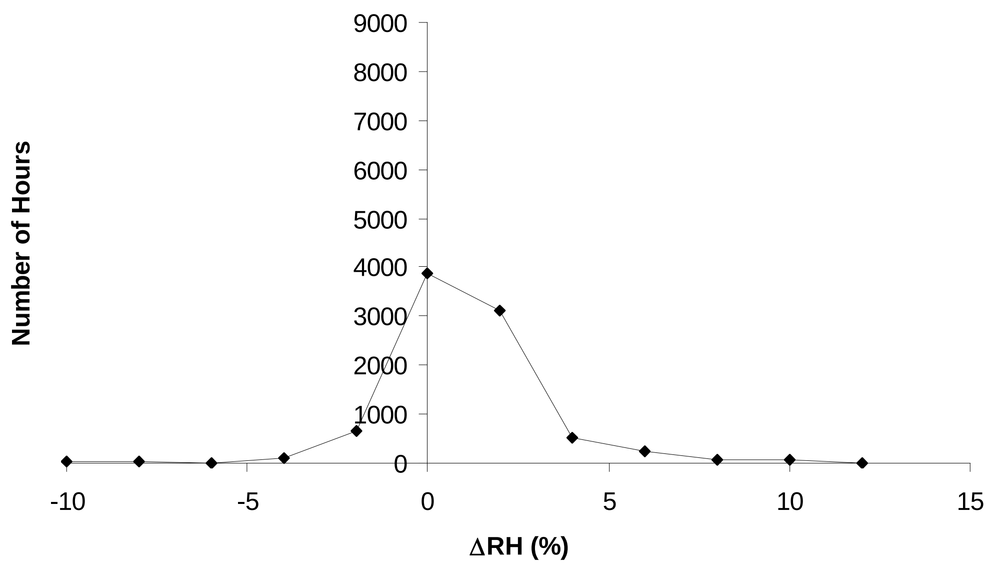

Figure 6.2. Relative humidity comparison for the fraction of recirculation air study in Saskatoon.

The annual consumption of natural gas and electricity are shown in Figure 6.3 for the 70% and 80% recirculation air cases. The electricity required for cooling is also shown separately. There is a significant reduction in the total amount of natural gas and electricity consumed in the 70% case as compared to the 80% case, while the electricity consumption for cooling is greater in the 70% recirculation case. By lowering the amount of recirculation air, the total air flow rate through the system is also lowered. A lower flow rate requires less power from the fans, which reduces the electrical energy consumption

It should be noted that in all sensitivity studies in this chapter, the bypass and economizer are controlled based on a constant design supply temperature of 14°C. In the sensitivity study on recirculation air, the supply air flow rate to the space is lower with 70% recirculation than with 80% recirculation. Therefore during part load operating conditions, the supply temperature would typically be lower in the 70% recirculation case than in the 80% recirculation case. As a result, the supply air will be overheated during some part load operating conditions. This could account for the large decrease in natural gas consumption and increase in cooling energy consumption. The increase in cooling, which dehumidifies the air would also account for the difference in relative humidity levels between the two cases.

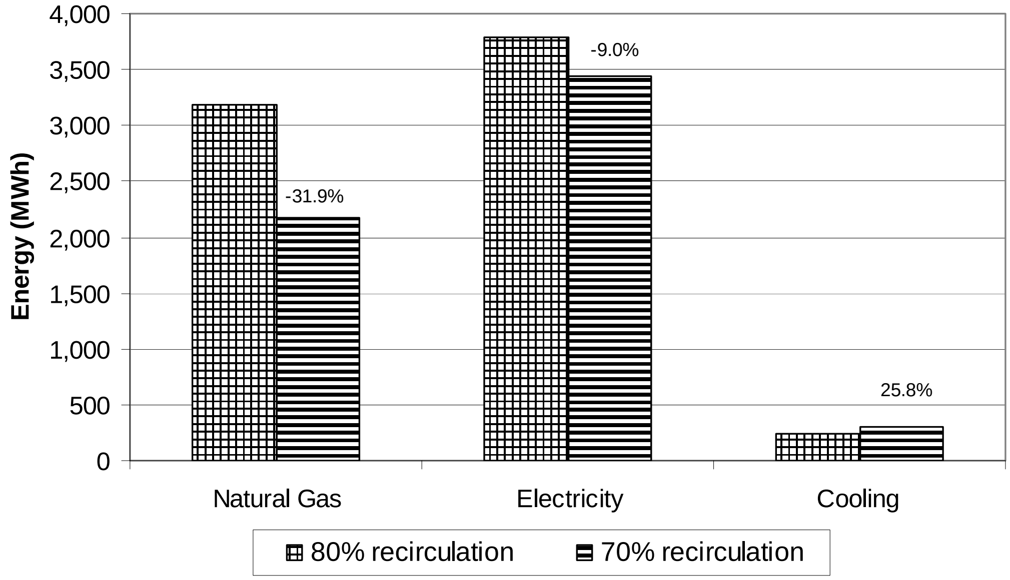

Figure 6.3. Energy consumption comparison for the fraction of recirculation air study in Saskatoon.

The equipment capacities for both the 70% recirculation air case and the 80% recirculation air case are shown in Figure 6.4. There is a decrease of 29% in the required boiler size in the 70% case, as compared to the 80% case. There is a 3% increase in the required size of the cooling coil for the 70% case. This again shows that the results of the simulation are sensitive to the fraction of recirculation air that is used in the system. It should be noted that the heating supply temperature was set at 30°C for both the 80% and 70% recirculation cases. Since the supply air flow rate is lower in the 70% recirculation case than in the 80% recirculation case, the result is that the boiler capacity is lower in the 70% recirculation case.

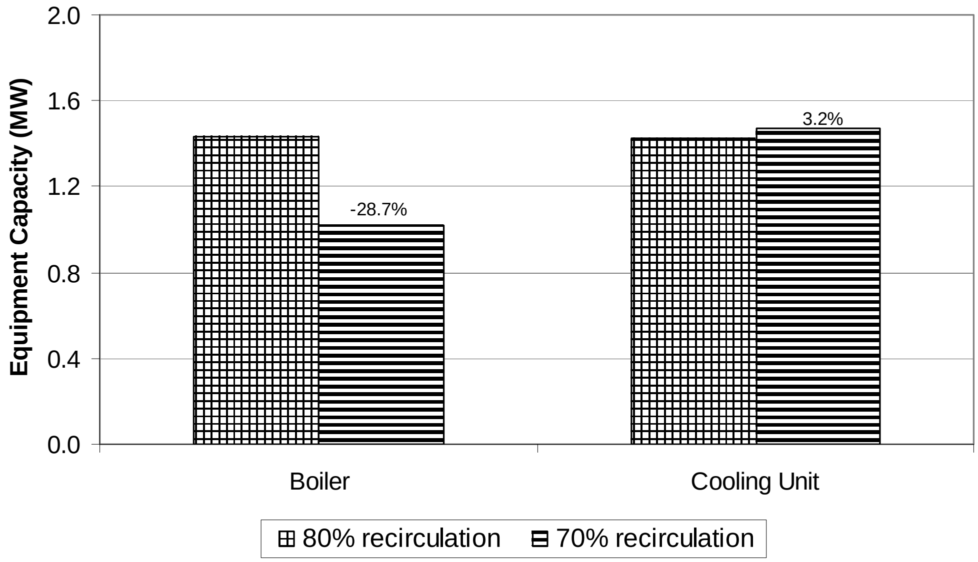

Figure 6.4. Equipment size comparison for the fraction of recirculation air study in Saskatoon.

6.2 SENSITIVITY OF THE ENERGY WHEEL EFFECTIVENESS

The effect of the energy wheel effectiveness (60% to 80%) on the indoor temperature and humidity are shown in Figure 6.5 and 6.6 respectively. As before, ΔT and ΔRH are the differences between the constant effectiveness energy wheel case and the variable effectiveness energy wheel case. Figures 6.5 and 6.6 show that the effectiveness of the energy wheel has a very small impact on the indoor temperature and humidity.

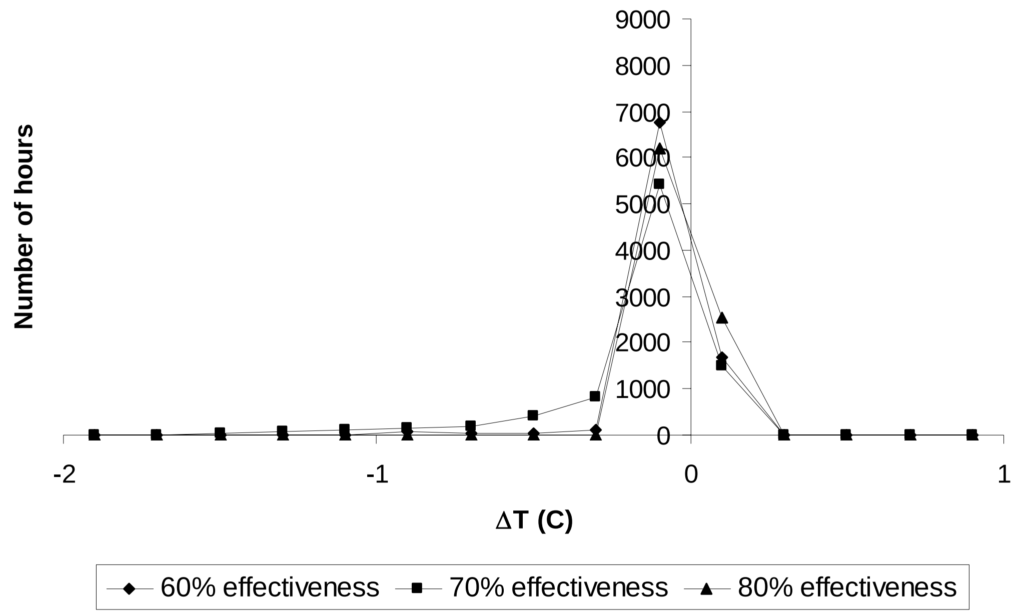

Figure 6.5. Temperature comparison for the energy wheel effectiveness study in Saskatoon.

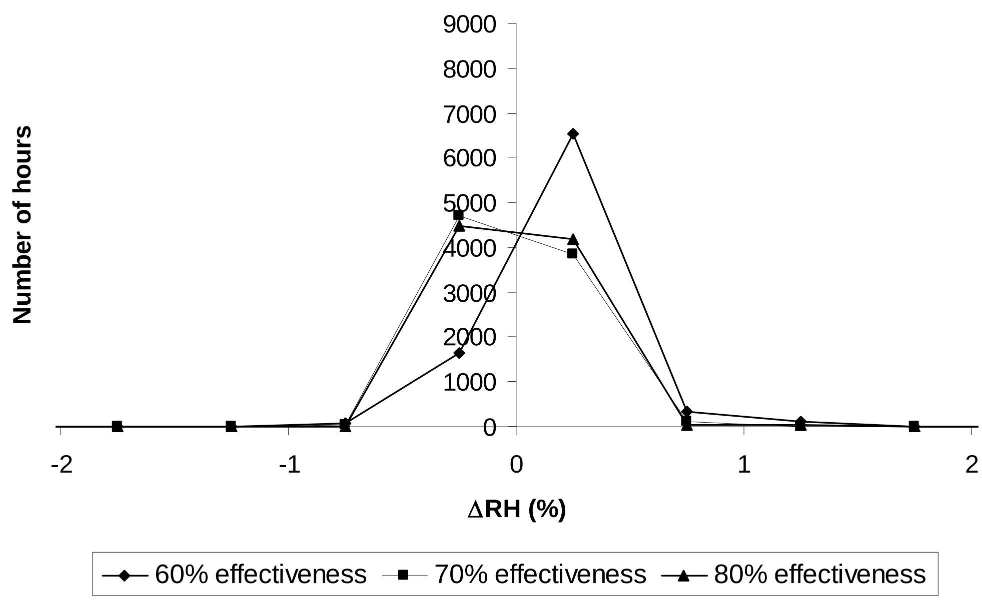

Figure 6.6. Relative humidity comparison for the energy wheel effectiveness study in Saskatoon.

Figure 6.7 shows the total natural gas and electricity consumption, as well as the electricity consumed for cooling, when the effectiveness (ε) ranges from 60% to 80%. Compared to the case with ε = 70%, there is a small reduction in the amount of natural gas and electricity consumed when ε = 80%. There is a small increase in natural gas consumption when ε = 60% as expected, and there is also a very small decrease in electricity consumption. A similar trend is seen for the equipment capacities as shown in Figure 6.8. It is expected that the cooling energy consumption and the cooling unit capacity would increase when the effectiveness of the wheel is reduced to 60% from 70%, which is not the result here. This is due to the constant control of the bypass/economizer.

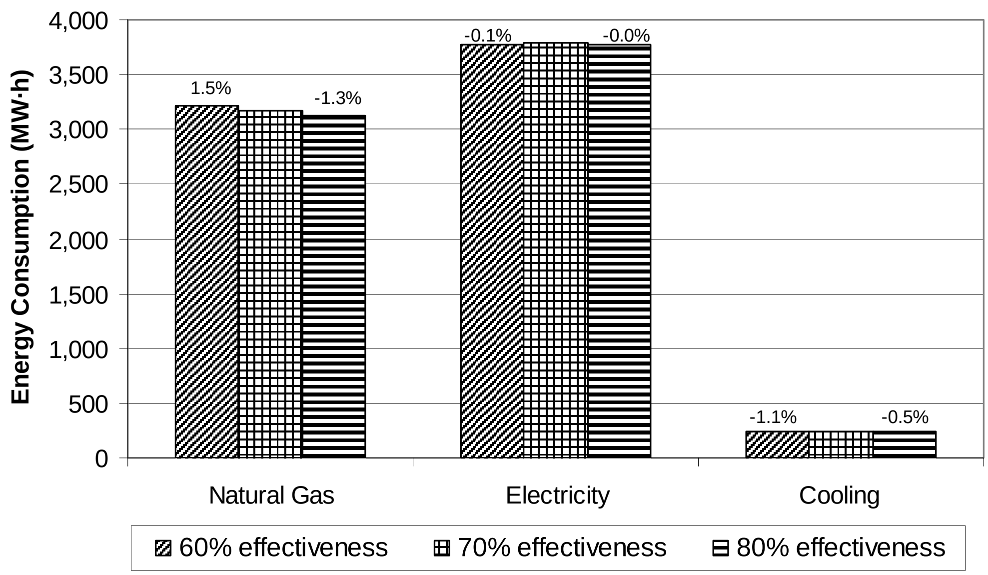

Figure 6.7. Energy consumption comparison for the energy wheel effectiveness study in Saskatoon.

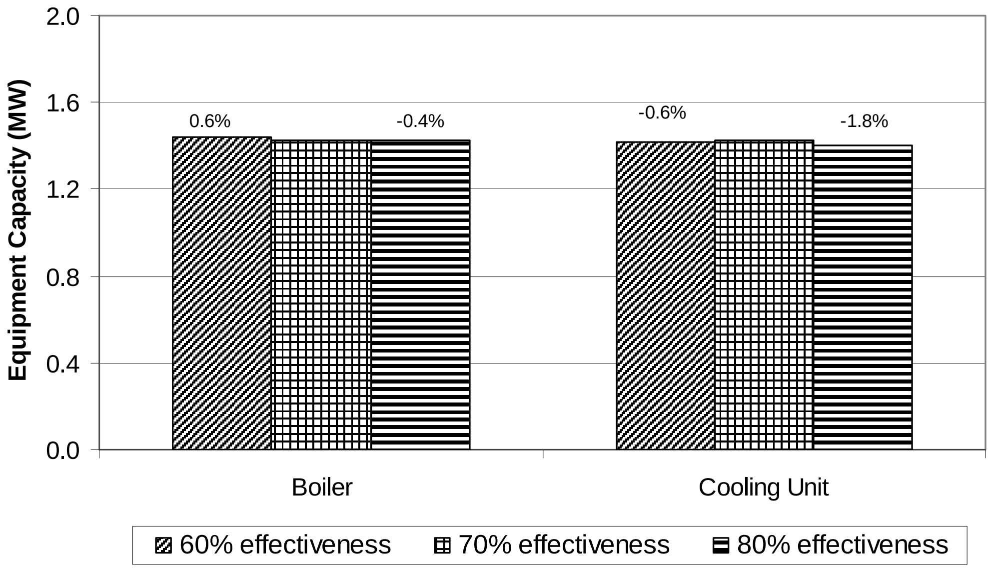

Figure 6.8. Equipment size comparison for the energy wheel effectiveness study in Saskatoon.

6.3 SENSITIVITY OF THE BUILDING CONSTRUCTION

The office and school buildings are both constructed with a typical building envelope for Saskatoon. This construction includes 7.5 cm of insulation in the walls and 5 cm of insulation in the roof. This building however, would not be a typical building in Tampa, or Phoenix. A building in Tampa is likely to have less insulation as the temperature there is quite warm during most of the year. To determine the effect this would have on the simulations a study is done on the office building in Tampa. The office buildings exterior construction is changed so that it has no insulation. In the following analysis, the building with no insulation is referred to as the typical construction, whereas the construction with insulation is referred to as the original construction. The difference between the indoor temperature calculated with the typical construction and the original construction is shown in Figure 6.9. The insulation has small effect on the indoor temperature as well as the indoor relative humidity (Figure 6.10).

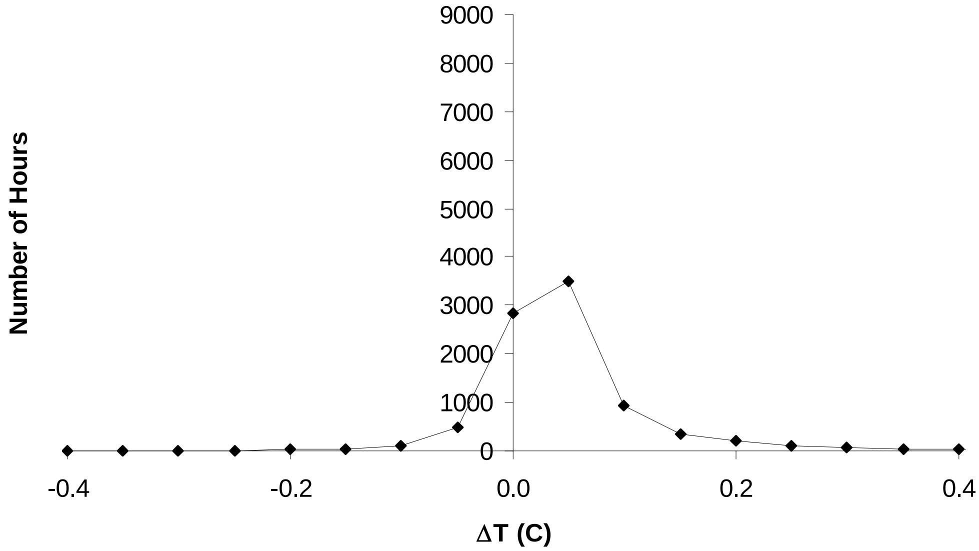

Figure 6.9. Temperature comparison for the building construction study in Tampa.

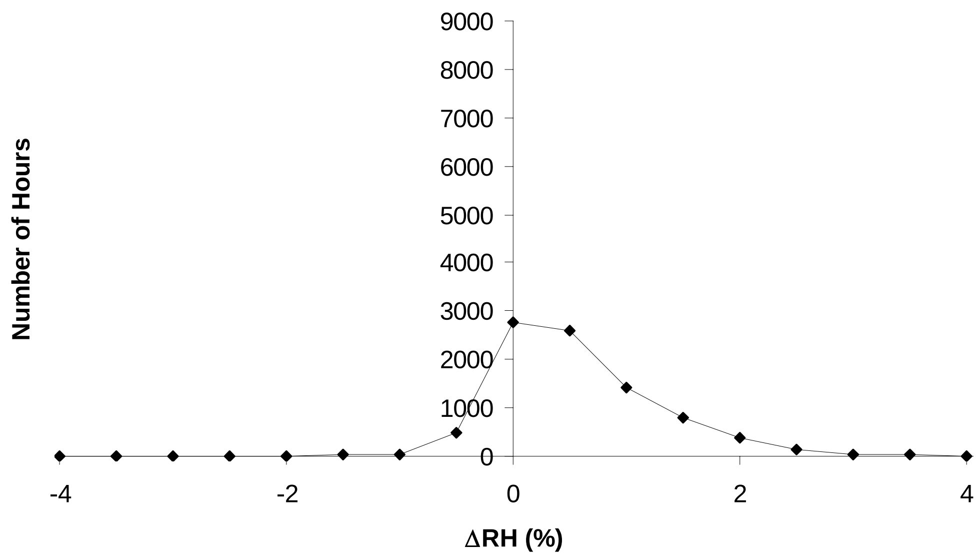

Figure 6.10. Relative humidity comparison for the building construction study in Tampa.

The annual energy consumption for the original and typical constructions is shown in Figure 6.11. There is an increase of 8% in the natural gas consumption from the original construction to the construction with no insulation. There is a small increase in the electricity consumed (0.9%) and in the electricity consumed due to the cooling load (2.6%) over the original construction case.

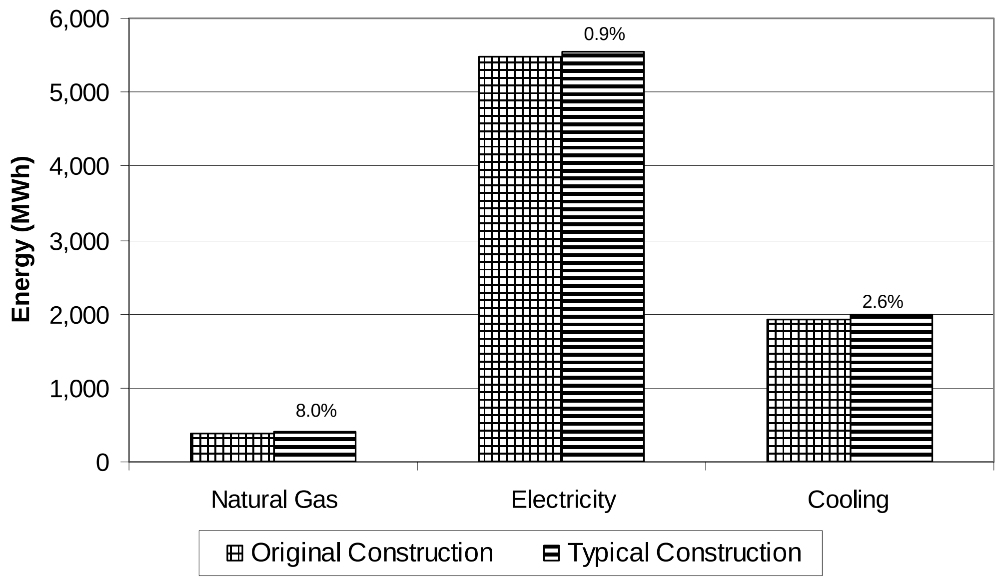

Figure 6.11. Energy consumption comparison for the building construction study in Tampa.

The size of the heating and cooling equipment for the original construction and the typical construction are shown in Figure 6.12. There is a considerable increase in the size of the heating equipment for the typical construction, compared to the original construction. This increase is almost 15%. There is also a small increase (2.2%) in the size of the cooling coil for the typical construction as expected.

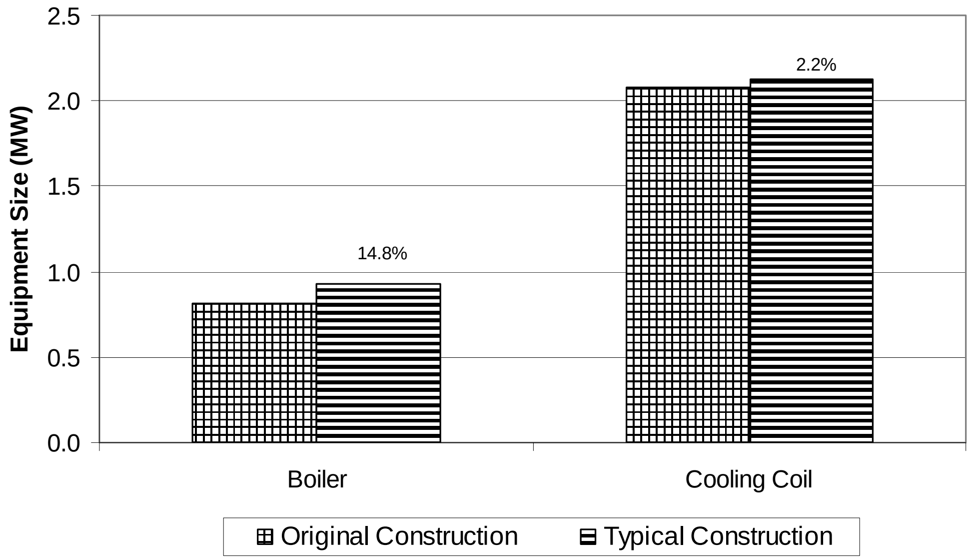

Figure 6.12. Equipment size comparison for the building construction study in Tampa.

6.4 SENSITIVITY OF THE VENTILATION RATES

The ventilation rate for each space is taken from ASHRAE Standard 62 (2001). There has been a new version of this standard published recently, ASHRAE Standard 62 (2004). The new standard is made up of two components, an occupant component and a floor area component. The total ventilation required is found by multiplying the occupant factor (R_p) by the number of people in the space and adding to that the product of the area factor (R_a) and the floor area as follows

$$Q_{vent} = R_p * P + R_a * A, \tag{6.1}$$

where P is the number of people and A is the floor area in m^2. Table 6.1 shows the values of R_p and R_a that are used to determine the new outdoor air requirements. The new ventilation standard gives a total ventilation rate of 11,100 L/s (23,500 cfm) which is 30% lower than the 2001 standard.

Table 6.1. Parameters required for the new ventilation standard including: number of people, floor area, occupant factor and area factor.

Space	People	Area (m^2)	R_p (L/(s·person))	R_a (L/(s·m^2))
Lobby	20	702	2.5	0.3
Washroom	14	174	-	-
Office	1523	21757.5	2.5	0.3
Elevator	-	13.4	-	0.3
Stairs	-	69	-	0.3

The temperature differences between the original simulations and the new ventilation simulations are shown in Figure 6.13. The difference in temperatures ranges from -1°C to 1°C.

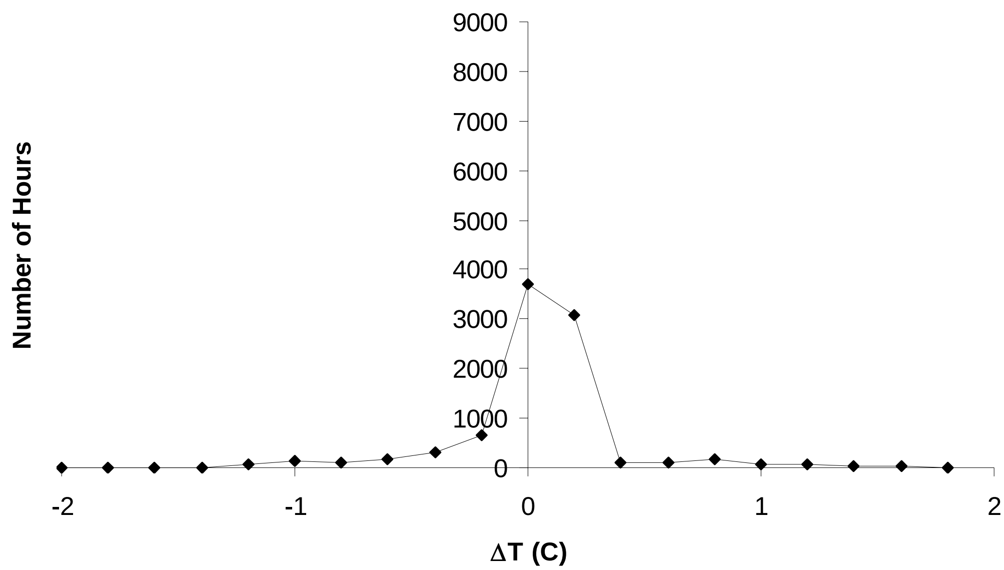

Figure 6.13. Temperature comparison for the ventilation rates study in Saskatoon.

The difference in relative humidity between the original simulations and the new ventilation simulations are shown in Figure 6.14. The difference in relative humidity varies from -5% RH to 10% RH. This is a significant change due to the change in ventilation rates.

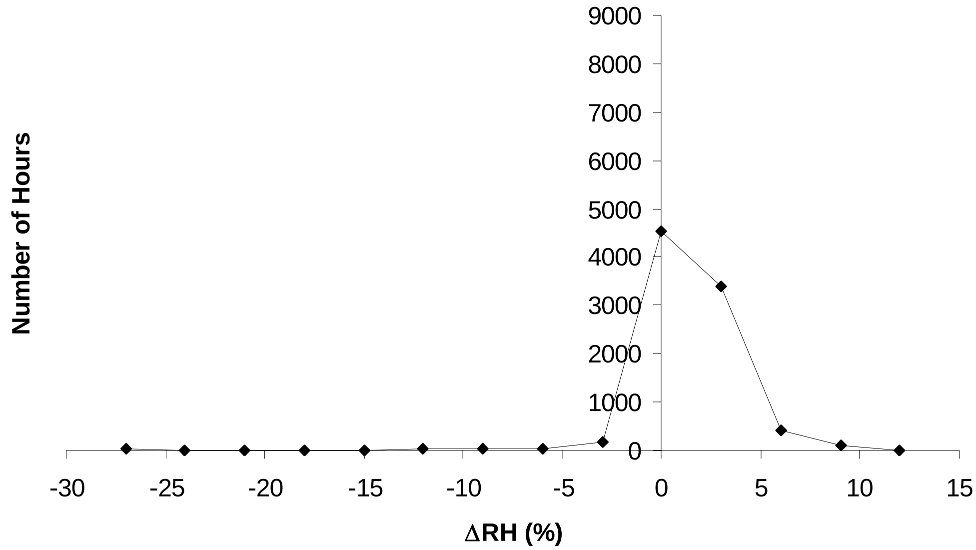

Figure 6.14. Relative humidity comparison for the ventilation rates study in Saskatoon.

Figure 6.15 shows the annual energy consumption when the ventilation rate is set according to the 2004 ventilation standard compared to the 2001 standard. There is a large decrease in the natural gas consumption when the 2004 ventilation standard is applied. There is also a significant reduction in the electricity consumed, although the actual electricity due to cooling increases with the new ventilation standard.

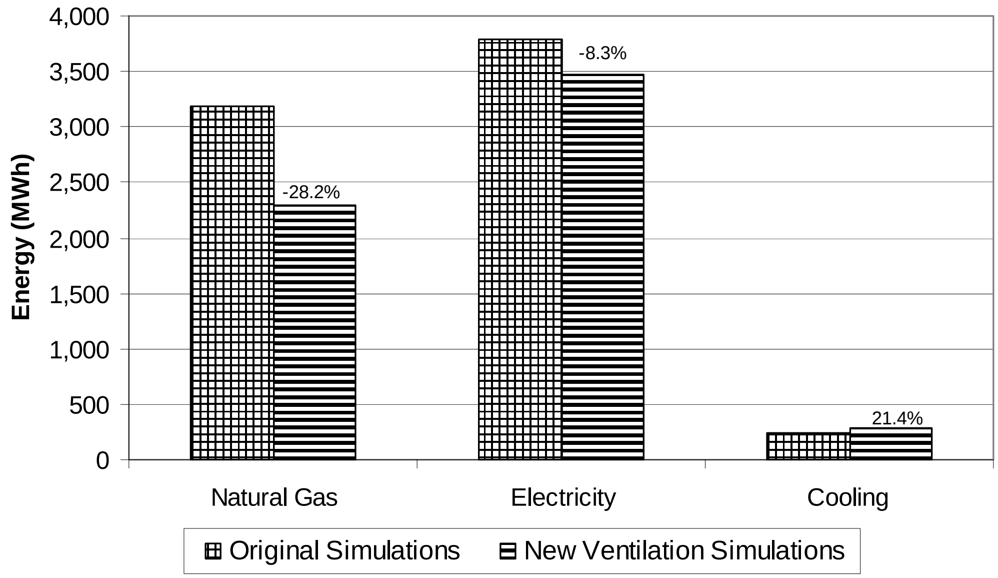

Figure 6.15. Energy consumption comparison for the ventilation rates study in Saskatoon.

The size of heating and cooling equipment is shown in Figure 6.16 for the original simulations and the new ventilation simulations. There is a large decrease in the size of the boiler when the new ventilation standard is used. There is a small increase in the size of the cooling coil with the use of the new ventilation standard. The results from the overall energy consumption and from the equipment size show that the new ventilation standard has a positive impact on the building simulations. When lowering the ventilation rate, it must be noted that there is a minimum ventilation rate for health that must be maintained. It should also be noted that the constant bypass/economizer control results in some overheating with the lower ventilation rate resulting in the increase in cooling energy consumption and the reduction in relative humidity between the two cases.

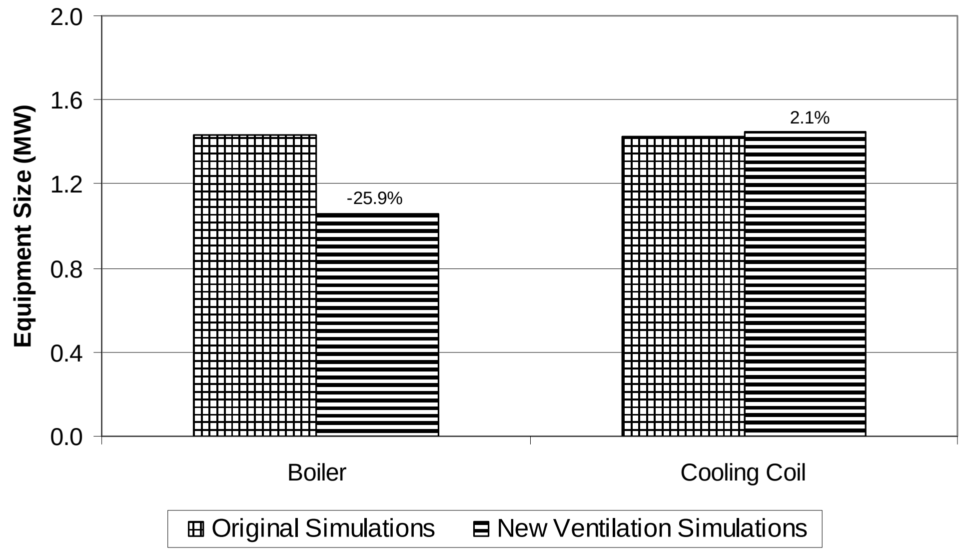

Figure 6.16. Equipment size comparison for the ventilation rates study in Saskatoon.

CHAPTER 7

CONCLUSIONS

In this thesis, the TRNSYS computer program was used to perform simulations on two buildings, an office and a school, in four different cities. These simulations were performed with and without an energy wheel to determine the effect of energy recovery on the indoor relative humidity (RH), perceived air quality (PAQ), thermal comfort and energy consumption.

7.1 CONCLUSIONS

The energy wheel is studied with both a constant effectiveness and a variable effectiveness. It is determined that the variable effectiveness has a negligible impact on the indoor relative humidity and the energy savings compared to the constant effectiveness. For this reason, it is acceptable to assume a constant effectiveness when simulating an energy wheel in an HVAC system.

The addition of an energy wheel into the HVAC system of a building is shown to have a large impact on the indoor RH and percent dissatisfied with PAQ in a humid climate, such as Tampa. In drier climates, such as Phoenix and Saskatoon, there is also a reduction in the indoor RH and percent dissatisfied with PAQ when the energy wheel is used, but it is not as large as in the humid climate. For a mild climate, such as

Vancouver, there is no significant change in the indoor RH or percent dissatisfied with PAQ. From this it is determined that an energy wheel can help improve the indoor climate and the perception of the indoor air quality in humid and dry climates.

The percent dissatisfied with thermal comfort is not reduced significantly with the use of an energy wheel, mainly due to the fact that the indoor temperature did not vary between the simulations with and without the energy wheel.

The amount of natural gas consumed for heating the building is reduced significantly in Saskatoon when the energy wheel is added to the HVAC system. The capacity of the boiler is also reduced, showing that an energy wheel can have a significant impact in a cold climate. The amount of electricity consumed for cooling and fans is reduced in Tampa and Phoenix with the addition of an energy wheel. As well, the capacity of the cooling unit is significantly reduced in each of these cities. This shows that the energy wheel can also have a significant effect in a hot climate. In Vancouver, however, there are no significant changes in either the energy consumption, or the capacity of the heating and cooling equipment with the energy wheel.

From the cost analysis, it is determined that an energy wheel would be economically feasible in Saskatoon, Tampa and Phoenix. By calculating the life-cycle cost of the HVAC equipment it is seen that a new building in these cities would experience savings when an energy wheel is added to the HVAC system, due to the reduction in the required capacities of the heating and cooling equipment, and reduced energy consumption. As well, the short payback periods (0-1 years for the office and 0-4.5 years for the school) in these three cities are very attractive. In Vancouver, there is actually an increase in the life-cycle cost of the HVAC system when an energy wheel is used, equal to the capital cost of the energy wheel. The payback period in Vancouver is very large and not feasible.

The sensitivity studies show that the simulation results are quite sensitive to the amount of recirculation air and the ventilation rate. When the amount of recirculation is lowered, there is a significant decrease in the indoor relative humidity and the energy consumption. Reducing the ventilation rate according to the new ASHRAE Standard (2004) also reduces the indoor relative humidity and energy consumption significantly. Changing the energy wheel effectiveness has a much smaller impact on the simulation results. A slight increase in energy consumption is seen when changing from 70% to 60% and there is a slight decrease in energy consumption when changing from 70% to 80%, as expected. The effectiveness of the wheel had a negligible effect on the indoor RH and PAQ. Finally, altering the building construction by removing the insulation has a positive effect on the indoor relative humidity, but a negative effect on the energy consumption. From these results it can be seen that care should be taken when selecting the recirculation and ventilation rates, and the building construction when designing buildings and estimating their indoor comfort and air quality conditions and energy consumption through simulation.

7.2 RECOMMENDATIONS FOR FUTURE WORK

The following recommendations are made for future work:

- perform a life cycle cost analysis including the cost of productivity in the office building,
- study how much the ventilation air flow rate can be increased in a building with an energy wheel without increasing energy consumption compared to a building without an energy wheel,
- perform a life cycle cost analysis of the HVAC system for different ventilation rates to determine the optimum ventilation rate for the buildings with and without an energy wheel.

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APPENDIX A

OFFICE BUILDING FILE CREATED IN PREBID

Office Building File Created in PREBID

```
*  PreBid   5.0.8
*   BUILDING DESCRIPTIONS FILE TRNSYS
*   FOR BUILDING:   C:\Thesiswork\Buildings\CV\Offices\Buildings\Office.bui
*   GET BY WORKING WITH PreBid 5.0 for Windows
*  C o m m e n t s
*  P r o j e c t
*+++ PROJECT
*+++ TITLE=BUI FOR TRNSYS 15.0
*+++ DESCRIPTION=GENERATED BY SIMCAD
*+++ CREATED=IISIBAT@CSTB.FR
*+++ ADDRESS=CSTB
*+++ CITY=F-06904 SOPHIA ANTIPOLIS
*+++ SWITCH=UNDEFINED
*  P r o p e r t i e s
PROPERTIES
 DENSITY=1.204 : CAPACITY=1.012 : HVAPOR=2454.0 : SIGMA=2.041e-007 : RTEMP=293.15
TYPES
*  L a y e r s
LAYER CONCRETE
 CONDUCTIVITY=   5.04 : CAPACITY=   0.88 : DENSITY=   2300
LAYER POLYINSUL
 CONDUCTIVITY= 0.0972 : CAPACITY=   1.21 : DENSITY=     55
LAYER ACCOUSTIC
 CONDUCTIVITY= 0.2088 : CAPACITY=   1.34 : DENSITY=    290
LAYER GYPSUM
 CONDUCTIVITY= 1.2672 : CAPACITY=   1.09 : DENSITY=    800
LAYER AIR
 CONDUCTIVITY=0.09468 : CAPACITY=   1.04 : DENSITY=   1.02
LAYER STEELBACK
 CONDUCTIVITY= 163.08 : CAPACITY=    0.5 : DENSITY=   7600
LAYER GLASSWALL
 CONDUCTIVITY= 204.48 : CAPACITY=   0.48 : DENSITY=   2230
LAYER BUILTUP
 CONDUCTIVITY=  0.612 : CAPACITY=   1.46 : DENSITY=   1100
LAYER GRAVEL
 CONDUCTIVITY=   2.52 : CAPACITY=      1 : DENSITY=   1800
*  I n p u t s
INPUTS LOBBYVENT WASHVENT OFFICEVENT ELEVATVENT STAIRSVENT TEMP RELHUM
*  S c h e d u l e s
SCHEDULE PEOPLE
 HOURS =0.0  8.0  11.0  12.0  13.0  14.0  17.0  18.0  21.0  24.0
 VALUES=0 1. 0.8 0.4 0.8 1. 0.3 0.1 0 0
SCHEDULE DAYWASH
 HOURS =0.0  8.0  18.0  24.0
 VALUES=0 1. 0 0
SCHEDULE COMPUTER
 HOURS =0.0  8.0  9.0  10.0  12.0  13.0  14.0  16.0  17.0  19.0  21.0  24.0
 VALUES=0 0.8 0.9 0.95 0.7 0.8 0.9 0.95 0.5 0.4 0 0
SCHEDULE WEEKEND
 HOURS =0.0  24.0
 VALUES=0 0
SCHEDULE LIGHTS
 HOURS =0.0  8.0  21.0  24.0
 VALUES=0.1 1. 0.1 0.1
SCHEDULE OFFLIGHTS
```

```
 HOURS =0.0  8.0  10.0  12.0  16.0  17.0  18.0  19.0  20.0  21.0  24.0
 VALUES=0.1 0.9 0.95 0.8 0.95 0.8 0.7 0.6 0.4 0.1 0.1
SCHEDULE WEEKENDLIG
 HOURS =0.0  24.0
 VALUES=0.1 0.1
SCHEDULE INFILLOBBY
 HOURS =0.0  8.0  9.0  12.0  13.0  17.0  18.0  24.0
 VALUES=0.22 0.37 0.3 0.37 0.3 0.37 0.22 0.22
SCHEDULE WEEKENDINF
 HOURS =0.0  24.0
 VALUES=0.22 0.22
SCHEDULE WEEKPEOPLE
 DAYS=1 2 3 4 5 6 7
 HOURLY=PEOPLE PEOPLE PEOPLE PEOPLE PEOPLE WEEKEND WEEKEND
SCHEDULE WASHPEOPLE
 DAYS=1 2 3 4 5 6 7
 HOURLY=DAYWASH DAYWASH DAYWASH DAYWASH DAYWASH WEEKEND WEEKEND
SCHEDULE WEEKCOMP
 DAYS=1 2 3 4 5 6 7
 HOURLY=COMPUTER COMPUTER COMPUTER COMPUTER COMPUTER WEEKEND
WEEKEND
SCHEDULE WEEKLIGHTS
 DAYS=1 2 3 4 5 6 7
 HOURLY=LIGHTS LIGHTS LIGHTS LIGHTS LIGHTS WEEKENDLIG WEEKENDLIG
SCHEDULE LIGHTSOFF
 DAYS=1 2 3 4 5 6 7
 HOURLY=OFFLIGHTS OFFLIGHTS OFFLIGHTS OFFLIGHTS OFFLIGHTS WEEKENDLIG
WEEKENDLIG
SCHEDULE LOBBYINFIL
 DAYS=1 2 3 4 5 6 7
 HOURLY=INFILLOBBY INFILLOBBY INFILLOBBY INFILLOBBY INFILLOBBY WEEKENDINF
WEEKENDINF
*  W a l l s
WALL FLOOR
 LAYERS   = CONCRETE POLYINSUL
 THICKNESS= 0.2      0.05
 ABS-FRONT= 0.6   : ABS-BACK= 0.6
 HFRONT   = 11 : HBACK= 64
WALL CEILING
 LAYERS   = ACCOUSTIC
 THICKNESS= 0.02
 ABS-FRONT= 0.6   : ABS-BACK= 0.6
 HFRONT   = 11 : HBACK= 11
WALL EXTERNAL
 LAYERS   = GYPSUM AIR   STEELBACK POLYINSUL GLASSWALL
 THICKNESS= 0.02   0.05  0.01      0.075     0.006
 ABS-FRONT= 0.6   : ABS-BACK= 0.6
 HFRONT   = 11 : HBACK= 64
WALL INTERNAL
 LAYERS   = CONCRETE
 THICKNESS= 0.15
 ABS-FRONT= 0.6   : ABS-BACK= 0.6
 HFRONT   = 11 : HBACK= 11
WALL SUBFLOOR
 LAYERS   = CONCRETE
 THICKNESS= 0.1
```

```
 ABS-FRONT= 0.6   : ABS-BACK= 0.6
 HFRONT   = 11 : HBACK= 11
WALL ROOF
 LAYERS   = CONCRETE POLYINSUL BUILTUP GRAVEL
 THICKNESS= 0.1      0.05      0.01    0.025
 ABS-FRONT= 0.6   : ABS-BACK= 0.6
 HFRONT   = 11 : HBACK= 64
* W i n d o w s
WINDOW DOUBLE
 WINID=2001 : HINSIDE=11 : HOUTSIDE=64 : SLOPE=90 : SPACID=0 : WWID=0 : WHEIG=0 :
FFRAME=0.15 : UFRAME=8.17 : ABSFRAME=0.6 : RISHADE=0.6 : RESHADE=0 :
REFLISHADE=0 : REFLOSHADE=0.5 : CCISHADE=0.5
* D e f a u l t  G a i n s
GAIN PERS_ISO04
 CONVECTIVE=180 : RADIATIVE=90 : HUMIDITY=0.11
GAIN COMPUTER04
 CONVECTIVE=690 : RADIATIVE=138 : HUMIDITY=0
GAIN LIGHT05_01
 CONVECTIVE=19206.7 : RADIATIVE=28810.1 : HUMIDITY=0
GAIN PERS_ISO01
 CONVECTIVE=144 : RADIATIVE=72 : HUMIDITY=0.059
GAIN LIGHT05_02
 CONVECTIVE=4760.64 : RADIATIVE=7140.96 : HUMIDITY=0
GAIN LIGHT05_03
 CONVECTIVE=595285 : RADIATIVE=892928 : HUMIDITY=0
GAIN LIGHT05_05
 CONVECTIVE=366.62 : RADIATIVE=549.94 : HUMIDITY=0
GAIN LIGHT05_06
 CONVECTIVE=1887.84 : RADIATIVE=2831.76 : HUMIDITY=0
* O t h e r  G a i n s
* C o m f o r t
COMFORT LOBBY
 CLOTHING=1 : MET=1.7 : WORK=0 : VELOCITY=0.1
COMFORT OFFICE
 CLOTHING=1 : MET=1.2 : WORK=0 : VELOCITY=0.1
* I n f i l t r a t i o n
INFILTRATION LOBBY
 AIRCHANGE=SCHEDULE 1*LOBBYINFIL
INFILTRATION OFFICE
 AIRCHANGE=0.29
INFILTRATION PLENUM
 AIRCHANGE=0.29
INFILTRATION MECHANICAL
 AIRCHANGE=0.29
* V e n t i l a t i o n
VENTILATION LOBBY
 TEMPERATURE=INPUT 1*TEMP
 AIRCHANGE=INPUT 1*LOBBYVENT
 HUMIDITY=INPUT 1*RELHUM
VENTILATION WASHROOM
 TEMPERATURE=INPUT 1*TEMP
 AIRCHANGE=INPUT 1*WASHVENT
 HUMIDITY=INPUT 1*RELHUM
VENTILATION OFFICE
 TEMPERATURE=INPUT 1*TEMP
 AIRCHANGE=INPUT 1*OFFICEVENT
```

```
 HUMIDITY=INPUT 1*RELHUM
VENTILATION ELEVATOR
 TEMPERATURE=INPUT 1*TEMP
 AIRCHANGE=INPUT 1*ELEVATVENT
 HUMIDITY=INPUT 1*RELHUM
VENTILATION STAIRS
 TEMPERATURE=INPUT 1*TEMP
 AIRCHANGE=INPUT 1*STAIRSVENT
 HUMIDITY=INPUT 1*RELHUM
* C o o l i n g
* H e a t i n g
* Z o n e s
ZONES LOBBY WASHROOM OFFICE PLENUM ELEVATOR STAIRS DUCT MECHANICAL
* O r i e n t a t i o n s
ORIENTATIONS NORTH SOUTH EAST WEST HORIZONT
BUILDING
* Z o n e LOBBY / A i r n o d e LOBBY
ZONE LOBBY
AIRNODE LOBBY
WALL  =FLOOR     : SURF=  1 : AREA=      702 : INTERNAL
WALL  =CEILING    : SURF=  2 : AREA=       36 : ADJACENT=PLENUM : FRONT
WALL  =CEILING    : SURF=  3 : AREA=      666 : ADJACENT=PLENUM : FRONT
WALL  =EXTERNAL   : SURF=  4 : AREA=    38.55 : EXTERNAL : ORI=SOUTH : FSKY=0.5
WINDOW=DOUBLE     : SURF=  5 : AREA=    34.35 : EXTERNAL : ORI=SOUTH : FSKY=0.5
WALL  =INTERNAL   : SURF=  6 : AREA=    113.4 : ADJACENT=OFFICE : BACK
WALL  =INTERNAL   : SURF=  7 : AREA=      2.7 : ADJACENT=WASHROOM : FRONT
WALL  =INTERNAL   : SURF=  8 : AREA=    23.49 : ADJACENT=ELEVATOR : FRONT
WALL  =INTERNAL   : SURF=  9 : AREA=     6.21 : ADJACENT=STAIRS : FRONT
WALL  =EXTERNAL   : SURF= 10 : AREA=    38.55 : EXTERNAL : ORI=WEST : FSKY=0.5
WINDOW=DOUBLE     : SURF= 11 : AREA=    34.35 : EXTERNAL : ORI=WEST : FSKY=0.5
 REGIME
 GAIN        = PERS_ISO04 : SCALE= SCHEDULE 20*WEEKPEOPLE
 GAIN        = COMPUTER04 : SCALE= SCHEDULE 6*WEEKCOMP
 GAIN        = LIGHT05_01 : SCALE= SCHEDULE 1*WEEKLIGHTS
 COMFORT     = LOBBY
 INFILTRATION= LOBBY
 VENTILATION = LOBBY
 CAPACITANCE = 22744.8 : VOLUME= 1895.4  : TINITIAL= 22      : PHINITIAL= 50      : WCAPR=
1
* Z o n e WASHROOM / A i r n o d e WASHROOM
ZONE WASHROOM
AIRNODE WASHROOM
WALL  =SUBFLOOR   : SURF= 12 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 13 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =INTERNAL   : SURF= 14 : AREA=    151.2 : INTERNAL
WALL  =INTERNAL   : SURF= 15 : AREA=    313.2 : ADJACENT=OFFICE : FRONT
WALL  =INTERNAL   : SURF= 16 : AREA=     70.2 : ADJACENT=DUCT : BACK
WALL  =INTERNAL   : SURF= 17 : AREA=   228.69 : ADJACENT=ELEVATOR : BACK
WALL  =SUBFLOOR   : SURF= 18 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 19 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 20 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 21 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 22 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 23 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 24 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 25 : AREA=        6 : ADJACENT=PLENUM : FRONT
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WALL  =SUBFLOOR   : SURF= 26 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 27 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 28 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 29 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 30 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 31 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 32 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 33 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 34 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 35 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 36 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 37 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 38 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 39 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 40 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 41 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 42 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 43 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 44 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 45 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 46 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 47 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 48 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 49 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 50 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 51 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 52 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 53 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 54 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 55 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 56 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 57 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 58 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 59 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 60 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 61 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 62 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 63 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 64 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 65 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 66 : AREA=        6 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 67 : AREA=        6 : ADJACENT=PLENUM : FRONT
WALL  =FLOOR      : SURF= 68 : AREA=       18 : INTERNAL
WALL  =CEILING    : SURF= 69 : AREA=        9 : ADJACENT=PLENUM : FRONT
WALL  =INTERNAL   : SURF= 70 : AREA=      2.7 : ADJACENT=LOBBY : BACK
WALL  =CEILING    : SURF= 71 : AREA=        9 : ADJACENT=PLENUM : FRONT
WALL  =INTERNAL   : SURF= 72 : AREA=     6.21 : ADJACENT=STAIRS : FRONT
 REGIME
 GAIN       = PERS_ISO01 : SCALE= SCHEDULE 14*WASHPEOPLE
 GAIN       = LIGHT05_02 : SCALE= SCHEDULE 1*WEEKLIGHTS
 VENTILATION = WASHROOM
 CAPACITANCE = 563.76  : VOLUME= 469.8   : TINITIAL= 22      : PHINITIAL= 50      : WCAPR= 1
* Z o n e  OFFICE  /  A i r n o d e  OFFICE
ZONE OFFICE
AIRNODE OFFICE
WALL  =SUBFLOOR   : SURF= 73 : AREA=   1003.5 : ADJACENT=PLENUM : BACK
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WALL  =CEILING    : SURF= 74 : AREA=    1003.5 : ADJACENT=PLENUM : FRONT
WALL  =EXTERNAL   : SURF= 75 : AREA=       870 : EXTERNAL : ORI=SOUTH : FSKY=0.5
WINDOW=DOUBLE     : SURF= 76 : AREA=       588 : EXTERNAL : ORI=SOUTH : FSKY=0.5
WALL  =INTERNAL   : SURF= 77 : AREA=    305.37 : ADJACENT=ELEVATOR : BACK
WALL  =INTERNAL   : SURF= 78 : AREA=    156.33 : ADJACENT=STAIRS : BACK
WALL  =EXTERNAL   : SURF= 79 : AREA=     913.5 : EXTERNAL : ORI=NORTH : FSKY=0.5
WINDOW=DOUBLE     : SURF= 80 : AREA=     617.4 : EXTERNAL : ORI=NORTH : FSKY=0.5
WALL  =EXTERNAL   : SURF= 81 : AREA=       870 : EXTERNAL : ORI=WEST : FSKY=0.5
WINDOW=DOUBLE     : SURF= 82 : AREA=       588 : EXTERNAL : ORI=WEST : FSKY=0.5
WALL  =SUBFLOOR   : SURF= 83 : AREA=    600.75 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 84 : AREA=    600.75 : ADJACENT=PLENUM : FRONT
WALL  =EXTERNAL   : SURF= 85 : AREA=     913.5 : EXTERNAL : ORI=EAST : FSKY=0.5
WINDOW=DOUBLE     : SURF= 86 : AREA=     617.4 : EXTERNAL : ORI=EAST : FSKY=0.5
WALL  =INTERNAL   : SURF= 87 : AREA=     175.5 : ADJACENT=DUCT : BACK
WALL  =INTERNAL   : SURF= 88 : AREA=     313.2 : ADJACENT=WASHROOM : BACK
WALL  =SUBFLOOR   : SURF= 89 : AREA=    1003.5 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 90 : AREA=    1003.5 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 91 : AREA=    600.75 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 92 : AREA=    600.75 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 93 : AREA=    1003.5 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 94 : AREA=    1003.5 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 95 : AREA=    600.75 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 96 : AREA=    600.75 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 97 : AREA=    1003.5 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF= 98 : AREA=    1003.5 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF= 99 : AREA=    600.75 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=100 : AREA=    600.75 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=101 : AREA=    1003.5 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=102 : AREA=    1003.5 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=103 : AREA=    600.75 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=104 : AREA=    600.75 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=105 : AREA=    1003.5 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=106 : AREA=    1003.5 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=107 : AREA=    600.75 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=108 : AREA=    600.75 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=109 : AREA=    1003.5 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=110 : AREA=    1003.5 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=111 : AREA=    600.75 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=112 : AREA=    600.75 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=113 : AREA=    1003.5 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=114 : AREA=    1003.5 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=115 : AREA=    600.75 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=116 : AREA=    600.75 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=117 : AREA=    1003.5 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=118 : AREA=    1003.5 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=119 : AREA=    600.75 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=120 : AREA=    600.75 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=121 : AREA=    1003.5 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=122 : AREA=    1003.5 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=123 : AREA=    600.75 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=124 : AREA=    600.75 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=125 : AREA=    1003.5 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=126 : AREA=    1003.5 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=127 : AREA=    598.95 : ADJACENT=PLENUM : BACK
WALL  =SUBFLOOR   : SURF=128 : AREA=       1.8 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=129 : AREA=    600.75 : ADJACENT=PLENUM : FRONT
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WALL  =SUBFLOOR   : SURF=130 : AREA=    1003.5 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=131 : AREA=    1003.5 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=132 : AREA=    600.75 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=133 : AREA=    598.95 : ADJACENT=PLENUM : FRONT
WALL  =CEILING    : SURF=134 : AREA=       1.8 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=135 : AREA=    1003.5 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=136 : AREA=    1003.5 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=137 : AREA=    600.75 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=138 : AREA=    600.75 : ADJACENT=PLENUM : FRONT
WALL  =FLOOR      : SURF=139 : AREA=    902.25 : INTERNAL
WALL  =CEILING    : SURF=140 : AREA=     337.5 : ADJACENT=PLENUM : FRONT
WALL  =CEILING    : SURF=141 : AREA=    564.75 : ADJACENT=PLENUM : FRONT
WALL  =INTERNAL   : SURF=142 : AREA=     113.4 : ADJACENT=LOBBY : FRONT
 REGIME
 GAIN        = PERS_ISO04 : SCALE= SCHEDULE 1523*WEEKPEOPLE
 GAIN        = COMPUTER04 : SCALE= SCHEDULE 1537*WEEKCOMP
 GAIN        = LIGHT05_03 : SCALE= SCHEDULE 1*LIGHTSOFF
 COMFORT     = OFFICE
 INFILTRATION= OFFICE
 VENTILATION = OFFICE
 CAPACITANCE = 704942  : VOLUME= 58745.2 : TINITIAL= 22     : PHINITIAL= 50     : WCAPR=
1
*  Z o n e  PLENUM  /  A i r n o d e  PLENUM
ZONE PLENUM
AIRNODE PLENUM
WALL  =CEILING    : SURF=143 : AREA=    1003.5 : ADJACENT=OFFICE : BACK
WALL  =CEILING    : SURF=144 : AREA=       2.3 : ADJACENT=STAIRS : BACK
WALL  =SUBFLOOR   : SURF=145 : AREA=    1003.5 : ADJACENT=MECHANICAL : FRONT
WALL  =SUBFLOOR   : SURF=146 : AREA=       2.3 : ADJACENT=STAIRS : FRONT
WALL  =EXTERNAL   : SURF=147 : AREA=       567 : EXTERNAL : ORI=SOUTH : FSKY=0.5
WALL  =INTERNAL   : SURF=148 : AREA=     234.5 : ADJACENT=ELEVATOR : FRONT
WALL  =EXTERNAL   : SURF=149 : AREA=       567 : EXTERNAL : ORI=NORTH : FSKY=0.5
WALL  =EXTERNAL   : SURF=150 : AREA=       567 : EXTERNAL : ORI=WEST : FSKY=0.5
WALL  =CEILING    : SURF=151 : AREA=       2.3 : ADJACENT=STAIRS : BACK
WALL  =CEILING    : SURF=152 : AREA=    600.75 : ADJACENT=OFFICE : BACK
WALL  =CEILING    : SURF=153 : AREA=         6 : ADJACENT=WASHROOM : BACK
WALL  =CEILING    : SURF=154 : AREA=         6 : ADJACENT=WASHROOM : BACK
WALL  =SUBFLOOR   : SURF=155 : AREA=       2.3 : ADJACENT=STAIRS : FRONT
WALL  =SUBFLOOR   : SURF=156 : AREA=    612.75 : ADJACENT=MECHANICAL : FRONT
WALL  =EXTERNAL   : SURF=157 : AREA=       567 : EXTERNAL : ORI=EAST : FSKY=0.5
WALL  =INTERNAL   : SURF=158 : AREA=     120.9 : ADJACENT=DUCT : FRONT
WALL  =CEILING    : SURF=159 : AREA=    1003.5 : ADJACENT=OFFICE : BACK
WALL  =CEILING    : SURF=160 : AREA=       2.3 : ADJACENT=STAIRS : BACK
WALL  =SUBFLOOR   : SURF=161 : AREA=       2.3 : ADJACENT=STAIRS : FRONT
WALL  =SUBFLOOR   : SURF=162 : AREA=    1003.5 : ADJACENT=OFFICE : FRONT
WALL  =CEILING    : SURF=163 : AREA=         6 : ADJACENT=WASHROOM : BACK
WALL  =CEILING    : SURF=164 : AREA=         6 : ADJACENT=WASHROOM : BACK
WALL  =CEILING    : SURF=165 : AREA=       2.3 : ADJACENT=STAIRS : BACK
WALL  =CEILING    : SURF=166 : AREA=    600.75 : ADJACENT=OFFICE : BACK
WALL  =SUBFLOOR   : SURF=167 : AREA=       2.3 : ADJACENT=STAIRS : FRONT
WALL  =SUBFLOOR   : SURF=168 : AREA=         6 : ADJACENT=WASHROOM : FRONT
WALL  =SUBFLOOR   : SURF=169 : AREA=         6 : ADJACENT=WASHROOM : FRONT
WALL  =SUBFLOOR   : SURF=170 : AREA=    600.75 : ADJACENT=OFFICE : FRONT
WALL  =CEILING    : SURF=171 : AREA=    1003.5 : ADJACENT=OFFICE : BACK
WALL  =CEILING    : SURF=172 : AREA=       2.3 : ADJACENT=STAIRS : BACK
WALL  =SUBFLOOR   : SURF=173 : AREA=       2.3 : ADJACENT=STAIRS : FRONT
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WALL  =SUBFLOOR   : SURF=174 : AREA=    1003.5 : ADJACENT=OFFICE : FRONT
WALL  =CEILING    : SURF=175 : AREA=         6 : ADJACENT=WASHROOM : BACK
WALL  =CEILING    : SURF=176 : AREA=         6 : ADJACENT=WASHROOM : BACK
WALL  =CEILING    : SURF=177 : AREA=    600.75 : ADJACENT=OFFICE : BACK
WALL  =CEILING    : SURF=178 : AREA=       2.3 : ADJACENT=STAIRS : BACK
WALL  =SUBFLOOR   : SURF=179 : AREA=       2.3 : ADJACENT=STAIRS : FRONT
WALL  =SUBFLOOR   : SURF=180 : AREA=    600.75 : ADJACENT=OFFICE : FRONT
WALL  =SUBFLOOR   : SURF=181 : AREA=         6 : ADJACENT=WASHROOM : FRONT
WALL  =SUBFLOOR   : SURF=182 : AREA=         6 : ADJACENT=WASHROOM : FRONT
WALL  =CEILING    : SURF=183 : AREA=    1003.5 : ADJACENT=OFFICE : BACK
WALL  =CEILING    : SURF=184 : AREA=       2.3 : ADJACENT=STAIRS : BACK
WALL  =SUBFLOOR   : SURF=185 : AREA=    1003.5 : ADJACENT=OFFICE : FRONT
WALL  =SUBFLOOR   : SURF=186 : AREA=       2.3 : ADJACENT=STAIRS : FRONT
WALL  =CEILING    : SURF=187 : AREA=    600.75 : ADJACENT=OFFICE : BACK
WALL  =CEILING    : SURF=188 : AREA=         6 : ADJACENT=WASHROOM : BACK
WALL  =CEILING    : SURF=189 : AREA=         6 : ADJACENT=WASHROOM : BACK
WALL  =CEILING    : SURF=190 : AREA=       2.3 : ADJACENT=STAIRS : BACK
WALL  =SUBFLOOR   : SURF=191 : AREA=         6 : ADJACENT=WASHROOM : FRONT
WALL  =SUBFLOOR   : SURF=192 : AREA=       2.3 : ADJACENT=STAIRS : FRONT
WALL  =SUBFLOOR   : SURF=193 : AREA=         6 : ADJACENT=WASHROOM : FRONT
WALL  =SUBFLOOR   : SURF=194 : AREA=    600.75 : ADJACENT=OFFICE : FRONT
WALL  =CEILING    : SURF=195 : AREA=       2.3 : ADJACENT=STAIRS : BACK
WALL  =CEILING    : SURF=196 : AREA=    1003.5 : ADJACENT=OFFICE : BACK
WALL  =SUBFLOOR   : SURF=197 : AREA=    1003.5 : ADJACENT=OFFICE : FRONT
WALL  =SUBFLOOR   : SURF=198 : AREA=       2.3 : ADJACENT=STAIRS : FRONT
WALL  =CEILING    : SURF=199 : AREA=       2.3 : ADJACENT=STAIRS : BACK
WALL  =CEILING    : SURF=200 : AREA=    600.75 : ADJACENT=OFFICE : BACK
WALL  =CEILING    : SURF=201 : AREA=         6 : ADJACENT=WASHROOM : BACK
WALL  =CEILING    : SURF=202 : AREA=         6 : ADJACENT=WASHROOM : BACK
WALL  =SUBFLOOR   : SURF=203 : AREA=         6 : ADJACENT=WASHROOM : FRONT
WALL  =SUBFLOOR   : SURF=204 : AREA=         6 : ADJACENT=WASHROOM : FRONT
WALL  =SUBFLOOR   : SURF=205 : AREA=       2.3 : ADJACENT=STAIRS : FRONT
WALL  =SUBFLOOR   : SURF=206 : AREA=    600.75 : ADJACENT=OFFICE : FRONT
WALL  =CEILING    : SURF=207 : AREA=       2.3 : ADJACENT=STAIRS : BACK
WALL  =CEILING    : SURF=208 : AREA=    1003.5 : ADJACENT=OFFICE : BACK
WALL  =SUBFLOOR   : SURF=209 : AREA=    1003.5 : ADJACENT=OFFICE : FRONT
WALL  =SUBFLOOR   : SURF=210 : AREA=       2.3 : ADJACENT=STAIRS : FRONT
WALL  =CEILING    : SURF=211 : AREA=    600.75 : ADJACENT=OFFICE : BACK
WALL  =CEILING    : SURF=212 : AREA=         6 : ADJACENT=WASHROOM : BACK
WALL  =CEILING    : SURF=213 : AREA=         6 : ADJACENT=WASHROOM : BACK
WALL  =CEILING    : SURF=214 : AREA=       2.3 : ADJACENT=STAIRS : BACK
WALL  =SUBFLOOR   : SURF=215 : AREA=         6 : ADJACENT=WASHROOM : FRONT
WALL  =SUBFLOOR   : SURF=216 : AREA=    600.75 : ADJACENT=OFFICE : FRONT
WALL  =SUBFLOOR   : SURF=217 : AREA=         6 : ADJACENT=WASHROOM : FRONT
WALL  =SUBFLOOR   : SURF=218 : AREA=       2.3 : ADJACENT=STAIRS : FRONT
WALL  =CEILING    : SURF=219 : AREA=       2.3 : ADJACENT=STAIRS : BACK
WALL  =CEILING    : SURF=220 : AREA=    1003.5 : ADJACENT=OFFICE : BACK
WALL  =SUBFLOOR   : SURF=221 : AREA=       2.3 : ADJACENT=STAIRS : FRONT
WALL  =SUBFLOOR   : SURF=222 : AREA=    1003.5 : ADJACENT=OFFICE : FRONT
WALL  =CEILING    : SURF=223 : AREA=    600.75 : ADJACENT=OFFICE : BACK
WALL  =CEILING    : SURF=224 : AREA=         6 : ADJACENT=WASHROOM : BACK
WALL  =CEILING    : SURF=225 : AREA=       2.3 : ADJACENT=STAIRS : BACK
WALL  =CEILING    : SURF=226 : AREA=         6 : ADJACENT=WASHROOM : BACK
WALL  =SUBFLOOR   : SURF=227 : AREA=    600.75 : ADJACENT=OFFICE : FRONT
WALL  =SUBFLOOR   : SURF=228 : AREA=         6 : ADJACENT=WASHROOM : FRONT
WALL  =SUBFLOOR   : SURF=229 : AREA=         6 : ADJACENT=WASHROOM : FRONT
```

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WALL  =SUBFLOOR   : SURF=230 : AREA=      2.3 : ADJACENT=STAIRS : FRONT
WALL  =CEILING    : SURF=231 : AREA=   1003.5 : ADJACENT=OFFICE : BACK
WALL  =CEILING    : SURF=232 : AREA=      2.3 : ADJACENT=STAIRS : BACK
WALL  =SUBFLOOR   : SURF=233 : AREA=      2.3 : ADJACENT=STAIRS : FRONT
WALL  =SUBFLOOR   : SURF=234 : AREA=   1003.5 : ADJACENT=OFFICE : FRONT
WALL  =CEILING    : SURF=235 : AREA=        6 : ADJACENT=WASHROOM : BACK
WALL  =CEILING    : SURF=236 : AREA=   600.75 : ADJACENT=OFFICE : BACK
WALL  =CEILING    : SURF=237 : AREA=      2.3 : ADJACENT=STAIRS : BACK
WALL  =CEILING    : SURF=238 : AREA=        6 : ADJACENT=WASHROOM : BACK
WALL  =SUBFLOOR   : SURF=239 : AREA=   600.75 : ADJACENT=OFFICE : FRONT
WALL  =SUBFLOOR   : SURF=240 : AREA=        6 : ADJACENT=WASHROOM : FRONT
WALL  =SUBFLOOR   : SURF=241 : AREA=      2.3 : ADJACENT=STAIRS : FRONT
WALL  =SUBFLOOR   : SURF=242 : AREA=        6 : ADJACENT=WASHROOM : FRONT
WALL  =CEILING    : SURF=243 : AREA=      2.3 : ADJACENT=STAIRS : BACK
WALL  =CEILING    : SURF=244 : AREA=   1003.5 : ADJACENT=OFFICE : BACK
WALL  =SUBFLOOR   : SURF=245 : AREA=   1003.5 : ADJACENT=OFFICE : FRONT
WALL  =SUBFLOOR   : SURF=246 : AREA=      2.3 : ADJACENT=STAIRS : FRONT
WALL  =CEILING    : SURF=247 : AREA=   600.75 : ADJACENT=OFFICE : BACK
WALL  =CEILING    : SURF=248 : AREA=        6 : ADJACENT=WASHROOM : BACK
WALL  =CEILING    : SURF=249 : AREA=        6 : ADJACENT=WASHROOM : BACK
WALL  =CEILING    : SURF=250 : AREA=      2.3 : ADJACENT=STAIRS : BACK
WALL  =SUBFLOOR   : SURF=251 : AREA=        6 : ADJACENT=WASHROOM : FRONT
WALL  =SUBFLOOR   : SURF=252 : AREA=      2.3 : ADJACENT=STAIRS : FRONT
WALL  =SUBFLOOR   : SURF=253 : AREA=        6 : ADJACENT=WASHROOM : FRONT
WALL  =SUBFLOOR   : SURF=254 : AREA=   600.75 : ADJACENT=OFFICE : FRONT
WALL  =CEILING    : SURF=255 : AREA=   1003.5 : ADJACENT=OFFICE : BACK
WALL  =CEILING    : SURF=256 : AREA=      2.3 : ADJACENT=STAIRS : BACK
WALL  =SUBFLOOR   : SURF=257 : AREA=   1003.5 : ADJACENT=OFFICE : FRONT
WALL  =SUBFLOOR   : SURF=258 : AREA=      2.3 : ADJACENT=STAIRS : FRONT
WALL  =CEILING    : SURF=259 : AREA=   600.75 : ADJACENT=OFFICE : BACK
WALL  =CEILING    : SURF=260 : AREA=      2.3 : ADJACENT=STAIRS : BACK
WALL  =CEILING    : SURF=261 : AREA=        6 : ADJACENT=WASHROOM : BACK
WALL  =CEILING    : SURF=262 : AREA=        6 : ADJACENT=WASHROOM : BACK
WALL  =SUBFLOOR   : SURF=263 : AREA=   600.75 : ADJACENT=OFFICE : FRONT
WALL  =SUBFLOOR   : SURF=264 : AREA=        6 : ADJACENT=WASHROOM : FRONT
WALL  =SUBFLOOR   : SURF=265 : AREA=        6 : ADJACENT=WASHROOM : FRONT
WALL  =SUBFLOOR   : SURF=266 : AREA=      2.3 : ADJACENT=STAIRS : FRONT
WALL  =CEILING    : SURF=267 : AREA=   1003.5 : ADJACENT=OFFICE : BACK
WALL  =CEILING    : SURF=268 : AREA=      2.3 : ADJACENT=STAIRS : BACK
WALL  =SUBFLOOR   : SURF=269 : AREA=   1003.5 : ADJACENT=OFFICE : FRONT
WALL  =SUBFLOOR   : SURF=270 : AREA=      2.3 : ADJACENT=STAIRS : FRONT
WALL  =CEILING    : SURF=271 : AREA=   600.75 : ADJACENT=OFFICE : BACK
WALL  =CEILING    : SURF=272 : AREA=        6 : ADJACENT=WASHROOM : BACK
WALL  =CEILING    : SURF=273 : AREA=        6 : ADJACENT=WASHROOM : BACK
WALL  =CEILING    : SURF=274 : AREA=      2.3 : ADJACENT=STAIRS : BACK
WALL  =SUBFLOOR   : SURF=275 : AREA=        6 : ADJACENT=WASHROOM : FRONT
WALL  =SUBFLOOR   : SURF=276 : AREA=      2.3 : ADJACENT=STAIRS : FRONT
WALL  =SUBFLOOR   : SURF=277 : AREA=        6 : ADJACENT=WASHROOM : FRONT
WALL  =SUBFLOOR   : SURF=278 : AREA=   600.75 : ADJACENT=OFFICE : FRONT
WALL  =CEILING    : SURF=279 : AREA=      1.8 : ADJACENT=OFFICE : BACK
WALL  =CEILING    : SURF=280 : AREA=      2.3 : ADJACENT=STAIRS : BACK
WALL  =CEILING    : SURF=281 : AREA=   1003.5 : ADJACENT=OFFICE : BACK
WALL  =SUBFLOOR   : SURF=282 : AREA=      1.8 : ADJACENT=OFFICE : FRONT
WALL  =SUBFLOOR   : SURF=283 : AREA=   1003.5 : ADJACENT=OFFICE : FRONT
WALL  =SUBFLOOR   : SURF=284 : AREA=      2.3 : ADJACENT=STAIRS : FRONT
WALL  =CEILING    : SURF=285 : AREA=        6 : ADJACENT=WASHROOM : BACK
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WALL  =CEILING    : SURF=286 : AREA=      2.3 : ADJACENT=STAIRS : BACK
WALL  =CEILING    : SURF=287 : AREA=        6 : ADJACENT=WASHROOM : BACK
WALL  =CEILING    : SURF=288 : AREA=   598.95 : ADJACENT=OFFICE : BACK
WALL  =SUBFLOOR   : SURF=289 : AREA=        6 : ADJACENT=WASHROOM : FRONT
WALL  =SUBFLOOR   : SURF=290 : AREA=      2.3 : ADJACENT=STAIRS : FRONT
WALL  =SUBFLOOR   : SURF=291 : AREA=   598.95 : ADJACENT=OFFICE : FRONT
WALL  =SUBFLOOR   : SURF=292 : AREA=        6 : ADJACENT=WASHROOM : FRONT
WALL  =CEILING    : SURF=293 : AREA=      2.3 : ADJACENT=STAIRS : BACK
WALL  =CEILING    : SURF=294 : AREA=   1003.5 : ADJACENT=OFFICE : BACK
WALL  =SUBFLOOR   : SURF=295 : AREA=      2.3 : ADJACENT=STAIRS : FRONT
WALL  =SUBFLOOR   : SURF=296 : AREA=   1003.5 : ADJACENT=OFFICE : FRONT
WALL  =CEILING    : SURF=297 : AREA=        6 : ADJACENT=WASHROOM : BACK
WALL  =CEILING    : SURF=298 : AREA=        6 : ADJACENT=WASHROOM : BACK
WALL  =CEILING    : SURF=299 : AREA=   600.75 : ADJACENT=OFFICE : BACK
WALL  =CEILING    : SURF=300 : AREA=      2.3 : ADJACENT=STAIRS : BACK
WALL  =SUBFLOOR   : SURF=301 : AREA=   600.75 : ADJACENT=OFFICE : FRONT
WALL  =SUBFLOOR   : SURF=302 : AREA=        6 : ADJACENT=WASHROOM : FRONT
WALL  =SUBFLOOR   : SURF=303 : AREA=      2.3 : ADJACENT=STAIRS : FRONT
WALL  =SUBFLOOR   : SURF=304 : AREA=        6 : ADJACENT=WASHROOM : FRONT
WALL  =CEILING    : SURF=305 : AREA=      2.3 : ADJACENT=STAIRS : BACK
WALL  =CEILING    : SURF=306 : AREA=    337.5 : ADJACENT=OFFICE : BACK
WALL  =CEILING    : SURF=307 : AREA=      666 : ADJACENT=LOBBY : BACK
WALL  =SUBFLOOR   : SURF=308 : AREA=      2.3 : ADJACENT=STAIRS : FRONT
WALL  =SUBFLOOR   : SURF=309 : AREA=   1003.5 : ADJACENT=OFFICE : FRONT
WALL  =CEILING    : SURF=310 : AREA=       36 : ADJACENT=LOBBY : BACK
WALL  =CEILING    : SURF=311 : AREA=        9 : ADJACENT=WASHROOM : BACK
WALL  =CEILING    : SURF=312 : AREA=   564.75 : ADJACENT=OFFICE : BACK
WALL  =CEILING    : SURF=313 : AREA=      2.3 : ADJACENT=STAIRS : BACK
WALL  =CEILING    : SURF=314 : AREA=        9 : ADJACENT=WASHROOM : BACK
WALL  =SUBFLOOR   : SURF=315 : AREA=        6 : ADJACENT=WASHROOM : FRONT
WALL  =INTERNAL   : SURF=316 : AREA=        6 : ADJACENT=DUCT : FRONT
WALL  =SUBFLOOR   : SURF=317 : AREA=      2.3 : ADJACENT=STAIRS : FRONT
WALL  =SUBFLOOR   : SURF=318 : AREA=        6 : ADJACENT=WASHROOM : FRONT
WALL  =SUBFLOOR   : SURF=319 : AREA=   600.75 : ADJACENT=OFFICE : FRONT
 REGIME
 INFILTRATION= PLENUM
 CAPACITANCE = 27237.5 : VOLUME= 22697.9 : TINITIAL= 22     : PHINITIAL= 50     : WCAPR=
1
*  Z o n e  ELEVATOR  /  A i r n o d e  ELEVATOR
ZONE ELEVATOR
AIRNODE ELEVATOR
WALL  =SUBFLOOR   : SURF=320 : AREA=      6.7 : ADJACENT=MECHANICAL : FRONT
WALL  =SUBFLOOR   : SURF=321 : AREA=      6.7 : ADJACENT=MECHANICAL : FRONT
WALL  =INTERNAL   : SURF=322 : AREA=    234.5 : ADJACENT=PLENUM : BACK
WALL  =INTERNAL   : SURF=323 : AREA=    33.67 : ADJACENT=DUCT : FRONT
WALL  =INTERNAL   : SURF=324 : AREA=   305.37 : ADJACENT=OFFICE : FRONT
WALL  =INTERNAL   : SURF=325 : AREA=   228.69 : ADJACENT=WASHROOM : FRONT
WALL  =INTERNAL   : SURF=326 : AREA=     75.6 : ADJACENT=STAIRS : BACK
WALL  =FLOOR      : SURF=327 : AREA=     13.4 : INTERNAL
WALL  =INTERNAL   : SURF=328 : AREA=    23.49 : ADJACENT=LOBBY : BACK
 REGIME
 GAIN       = LIGHT05_05 : SCALE= SCHEDULE 1*WEEKLIGHTS
 VENTILATION = ELEVATOR
 CAPACITANCE = 832.94  : VOLUME= 694.12  : TINITIAL= 22     : PHINITIAL= 50     : WCAPR=
1
*  Z o n e  STAIRS  /  A i r n o d e  STAIRS
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```
ZONE STAIRS
AIRNODE STAIRS
WALL  =SUBFLOOR   : SURF=329 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =SUBFLOOR   : SURF=330 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =ROOF       : SURF=331 : AREA=      4.6 : EXTERNAL : ORI=HORIZONT : FSKY=1
WALL  =INTERNAL   : SURF=332 : AREA=    17.01 : ADJACENT=MECHANICAL : FRONT
WALL  =INTERNAL   : SURF=333 : AREA=    86.94 : ADJACENT=DUCT : FRONT
WALL  =SUBFLOOR   : SURF=334 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =SUBFLOOR   : SURF=335 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=336 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =CEILING    : SURF=337 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =INTERNAL   : SURF=338 : AREA=     75.6 : ADJACENT=ELEVATOR : FRONT
WALL  =INTERNAL   : SURF=339 : AREA=   156.33 : ADJACENT=OFFICE : FRONT
WALL  =SUBFLOOR   : SURF=340 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =SUBFLOOR   : SURF=341 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=342 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =CEILING    : SURF=343 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=344 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =SUBFLOOR   : SURF=345 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=346 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =CEILING    : SURF=347 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=348 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =SUBFLOOR   : SURF=349 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=350 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =CEILING    : SURF=351 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=352 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =SUBFLOOR   : SURF=353 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=354 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =CEILING    : SURF=355 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=356 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =SUBFLOOR   : SURF=357 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=358 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =CEILING    : SURF=359 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=360 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =SUBFLOOR   : SURF=361 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=362 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =CEILING    : SURF=363 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=364 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =SUBFLOOR   : SURF=365 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=366 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =CEILING    : SURF=367 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=368 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =SUBFLOOR   : SURF=369 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=370 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =CEILING    : SURF=371 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=372 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =SUBFLOOR   : SURF=373 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=374 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =CEILING    : SURF=375 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=376 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =SUBFLOOR   : SURF=377 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=378 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =CEILING    : SURF=379 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=380 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =SUBFLOOR   : SURF=381 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=382 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
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WALL  =CEILING    : SURF=383 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =SUBFLOOR   : SURF=384 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =SUBFLOOR   : SURF=385 : AREA=      2.3 : ADJACENT=PLENUM : BACK
WALL  =CEILING    : SURF=386 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =CEILING    : SURF=387 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =FLOOR      : SURF=388 : AREA=      4.6 : INTERNAL
WALL  =CEILING    : SURF=389 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =CEILING    : SURF=390 : AREA=      2.3 : ADJACENT=PLENUM : FRONT
WALL  =INTERNAL   : SURF=391 : AREA=     6.21 : ADJACENT=WASHROOM : BACK
WALL  =INTERNAL   : SURF=392 : AREA=     6.21 : ADJACENT=LOBBY : BACK
 REGIME
 GAIN        = LIGHT05_06 : SCALE= SCHEDULE 1*WEEKLIGHTS
 VENTILATION = STAIRS
 CAPACITANCE = 223.56  : VOLUME= 186.3   : TINITIAL= 22      : PHINITIAL= 50      : WCAPR= 1
*  Z o n e  DUCT  /  A i r n o d e  DUCT
ZONE DUCT
AIRNODE DUCT
WALL  =ROOF       : SURF=393 : AREA=        6 : EXTERNAL : ORI=HORIZONT : FSKY=1
WALL  =INTERNAL   : SURF=394 : AREA=    20.79 : ADJACENT=MECHANICAL : FRONT
WALL  =INTERNAL   : SURF=395 : AREA=    86.94 : ADJACENT=STAIRS : BACK
WALL  =INTERNAL   : SURF=396 : AREA=    120.9 : ADJACENT=PLENUM : BACK
WALL  =INTERNAL   : SURF=397 : AREA=    33.67 : ADJACENT=ELEVATOR : BACK
WALL  =INTERNAL   : SURF=398 : AREA=     70.2 : ADJACENT=WASHROOM : FRONT
WALL  =INTERNAL   : SURF=399 : AREA=    175.5 : ADJACENT=OFFICE : FRONT
WALL  =INTERNAL   : SURF=400 : AREA=        6 : ADJACENT=PLENUM : BACK
 REGIME
 CAPACITANCE = 365.76  : VOLUME= 304.8   : TINITIAL= 22      : PHINITIAL= 50      : WCAPR= 1
*  Z o n e  MECHANICAL  /  A i r n o d e  MECHANICAL
ZONE MECHANICAL
AIRNODE MECHANICAL
WALL  =SUBFLOOR   : SURF=401 : AREA=      6.7 : ADJACENT=ELEVATOR : BACK
WALL  =SUBFLOOR   : SURF=402 : AREA=   1003.5 : ADJACENT=PLENUM : BACK
WALL  =ROOF       : SURF=403 : AREA=  1629.65 : EXTERNAL : ORI=HORIZONT : FSKY=1
WALL  =EXTERNAL   : SURF=404 : AREA=   109.35 : EXTERNAL : ORI=SOUTH : FSKY=0.5
WALL  =INTERNAL   : SURF=405 : AREA=    17.01 : ADJACENT=STAIRS : BACK
WALL  =EXTERNAL   : SURF=406 : AREA=   109.35 : EXTERNAL : ORI=NORTH : FSKY=0.5
WALL  =EXTERNAL   : SURF=407 : AREA=   109.35 : EXTERNAL : ORI=WEST : FSKY=0.5
WALL  =SUBFLOOR   : SURF=408 : AREA=   612.75 : ADJACENT=PLENUM : BACK
WALL  =SUBFLOOR   : SURF=409 : AREA=      6.7 : ADJACENT=ELEVATOR : BACK
WALL  =EXTERNAL   : SURF=410 : AREA=   109.35 : EXTERNAL : ORI=EAST : FSKY=0.5
WALL  =INTERNAL   : SURF=411 : AREA=    20.79 : ADJACENT=DUCT : BACK
 REGIME
 INFILTRATION= MECHANICAL
 CAPACITANCE = 5280.07 : VOLUME= 4400.06 : TINITIAL= 22      : PHINITIAL= 50      : WCAPR=
1
*  O u t p u t s
OUTPUTS
 TRANSFER : TIMEBASE=1.000
 AIRNODES = LOBBY WASHROOM OFFICE PLENUM ELEVATOR STAIRS DUCT
MECHANICAL
 NTYPES =   1 : TAIR      air temperature of zone
        =   9 : RELHUM     relativ humidity of zone air
 AIRNODES = LOBBY WASHROOM OFFICE ELEVATOR STAIRS
 NTYPES =  29 : ABSHUM          absolute humidity of zone air
 AIRNODES = LOBBY OFFICE
 NTYPES =  62 : PMV      predicted mean vote (PMV) value of zone
```

= 63 : PPD predicted percentage of dissatisfied persons (PPD) of zone
* E n d
END

APPENDIX B

DETAILED SEARCH FOR A COMPUTER SIMULATION PACKAGE

Review of building energy simulation tools that include moisture storage in building materials and HVAC systems

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Abstract

This report is a first draft of a review of building energy software for IEA Annex 41. The purpose of this report is to make a preliminary assessment of the adaptability of building energy simulation tools to whole building heat, air and moisture (HAM) transfer as defined in Annex 41 including detailed indoor air models (e.g., air movement and moisture sources), outdoor weather models (e.g., wind driven rain and exterior condensation), envelope models (e.g., diffusion, convection) and HVAC models. Whole building HAM transfer is not possible with any of the reviewed models. Therefore, this report will attempt to identify those energy tools that include moisture storage in building materials and HVAC systems. Although most programs include thermal storage, only a few include moisture storage and transport in furniture and constructions. The most common hygroscopic material for HVAC systems is a rotating wheel coated with hygroscopic desiccants (energy wheel). This air-to-air energy exchanger allows the transfer of both heat and moisture. Again, it is common for energy tools to model sensible heat exchangers, but relatively few model moisture exchangers.

A search was performed and a list of over 260 possible programs was created. Once the list was complete, the review process began. Many programs were eliminated because they were too specific and did not take into account all the necessary components needed for a full energy analysis. Others lacked the essential information to decide whether they would be suitable for this research. After these programs were removed, there were 11 programs left for consideration. These programs were then researched further, and a representative of each company was contacted. Based on this information, more information is presented on four programs (TRNSYS, BSim, EnergyPlus and DOE) that appeared to be rather appropriate. It is important to note that this report is based on information from representatives of these building energy simulation tools and literature, and not from personal experience with the programs, except for demonstration versions of TRNSYS and BSim.

1.0 Introduction

The energy calculations that are required to analyze a building can be quite complex and laborious. An easier way to solve these equations is to use a building energy simulation tool. These computer programs can simulate a building and its HVAC system. They can predict results such as the heating and cooling loads of a building, as well as the indoor thermal climate of a space. There are many simulation packages available that can perform a number of different tasks. Each one varies in the way it takes input, the method it uses to calculate the results and the number of results that are obtainable. This report provides a preliminary review of programs that have a few specific desired qualities.

Before the software package can accurately simulate a building, there are some key aspects that need be entered into the program. These include the geometry of the building and the number of zones that make up the building, as well as the internal loads within the building. Internal loads include the latent and sensible heat given off by people, lighting, equipment and hygroscopic materials. One key aspect that will be focused on in this report is the inclusion of moisture storage, in addition to the typical thermal storage. Moisture storage, in materials within the building, as well as the building envelope, can have a significant effect on the indoor thermal climate of the building. Moisture storage is not a common feature in building energy tools and thus it will reduce the number of tools that need to be reviewed in detail.

Another way in which building energy tools vary is in the type of simulations that they can perform. Most programs can calculate the peak heating and cooling loads, the total energy consumption, system performance and costs. Many programs also estimate the indoor temperature, however, the capability to predict the indoor humidity is also required for the research in Annex 41. Relevant energy tools should also be able to simulate both heat and moisture transfer exchangers. While most programs can model sensible heat exchangers, many do not able to model moisture exchangers. Finally, the simulation package should allow the user to program and simulate components and systems that are not available or inaccurately modeled in the tool. The user should therefore have access to the source code to allow new research findings to be included in the program.

2.0 Building Energy Simulation Tools

Using the Building Energy Software Tools Directory (United States Department of Energy, 2003), a website listing 265 different energy-related software tools, a list was made of all software packages that might have the desired capabilities. A search of the internet was performed on all selected tools, as well as for any other simulation tools that might be suitable. This search was performed in order to determine what kind of simulations each tool could perform, as well as what data were required for input into the program. A literature search was also performed to find any additional information that might be available. After this, the list was narrowed down to those programs that might have the best chance of simulating whole building HAM transfer and energy consumption. Finally, the companies that made each program were contacted by email to find out more information on the building energy tools.

After the internet search was performed, there remained only 11 computer software packages that seemed suitable. These programs were BLAST, BSim, CHVAC, DOE-2, EnergyPlus, HAP, IDA ICE, NewQUICK, RIUSKA, TRNSYS and UMIDUS. In the following section, each of these computer packages will be introduced in alphabetical order, and a brief description of its capabilities given. The information in each of the summaries was obtained via the internet, as well as from personal communication with the company that created the software. Finally, a summary of the capabilities of the 11 tools will be presented. It should be noted that the authors of this report have not used the simulation tools, but have relied on information from suppliers and the literature.

2.1 BLAST

The Building Loads Analysis and System Thermodynamics (BLAST) (Al-Rabghi and Hittle, 2001, and Lam and Chan, 2001) program was sponsored by the Engineering Departments within the U.S. Air Force and the U.S. Army. It is now supported and maintained by the Building Systems Laboratory at the University of Illinois at Urbana-Champaign (2003). BLAST provides a simulation of a building's energy needs, including the energy consumption, system performance and cost. As well, it calculates hourly heating and cooling loads based on a fundamental heat balance method. The program can also include air-to-air heat recovery for most systems. Information on the building structure can be input from a library which has data from ASHRAE Fundamentals. This includes materials, and wall, roof and floor sections.

Some advantages of this program are that it has a short run time, and that it uses detailed algorithms for calculations. The use of detailed algorithms allows the program to calculate more parameters, ones that might not be available to other programs. A downside to this program is that it requires a high level of expertise. Although the program does have some good qualities, its biggest disadvantage was that it would not have the desired moisture related capabilities. Therefore, this program was not investigated further in this report.

2.2 BSim

The BSim family of computer simulation tools was created by Danish Building and Urban Research (2003). This program provides an analysis of the indoor climate, moisture conditions, energy consumption and daylight performance of the building. The latest model, BSim2002 (Rode et al., 2001 and 2003), has a model for both heat and moisture recovery. The package also contains a database with moisture properties of different materials that can be input into the program. This simulates the moisture within the building construction. It also allows the user to follow the effects that the moisture has on the indoor climate and the moisture conditions throughout the different layers of construction. This building simulation tool has the ability to analyze complex buildings, as well as those with special requirements for the indoor climate. According to the company website, there is an extension module for BSim2002 that can perform advanced moisture simulations. Since BSim meets the criteria, it will be evaluated further in section 3 of this report.

2.3 CHVAC

CHVAC is a product of Elite Software (2003). It is a simulation tool for calculating peak heating and cooling loads for commercial buildings. There is also a related simulation tool available for residential buildings, called RHVAC. The building loads are calculated using the somewhat outdated CLTD procedures as described in the 1997 ASHRAE Handbook of Fundamentals. This method is not included in current editions. This allows the program to perform calculations for buildings of any orientation. The building to be analyzed can be broken down into different zones, the number of which depends on the package purchased. There are five different versions, consisting of 2 zones, 10 zones, 50 zones, 100 zones and an unlimited number of zones. Of course, with the increase in the number of zones available, there is also an increase in price. According to Elite Software, the CHVAC program is meant to only calculate design loads and not perform an energy analysis on the building. It is also incapable of incorporating moisture storage into the simulation. Elite Software does however have another program called EZDOE which can perform an energy balance on the building; unfortunately, it does not include moisture storage either. These programs are not considered further.

2.4 DOE-2

The DOE-2 (Building Energy Analysis Simulation Research Group, 2003, Kony and Kossecka, 2002 and Carriere et al., 1999) building simulation package was developed by the Department of Energy with the intention of performing whole building energy analyses of both commercial and residential buildings. The program simulates the energy performance of the building, including the life cycle cost of operating the HVAC system. Within the DOE-2 program, there are four subprograms: Loads, System, Plant and Economics. The building geometry and other input data are given to the loads program and the hourly heating and cooling loads are returned. These loads are then fed into the system program which determines the amount of energy required to condition the air to meet these loads. The plant program then models the HVAC equipment necessary to meet desired conditions. Finally, the Economics program calculates the cost of energy used by the HVAC system.

DOE-2 has been extensively validated by comparison with actual measurements and calculations. For this reason, DOE-2 comes highly recommended by ASHRAE as a complete energy simulation tool. The equations used to perform all calculations within the program are based on ASHRAE published algorithms. While the program comes highly praised, there are a few disadvantages of the DOE-2 program. The input into the program can be time consuming and the level of knowledge required from the user is high. As well, it does not have the ability to simulate some things, such as IAQ. After conferring with the Department of Energy, it is still not certain whether the program can simulate moisture storage and air-to-air energy recovery as part of its built-in functions. Air-to-air energy recovery can be simulated through a user programmed subroutine, but moisture storage is still not likely possible. Nevertheless, this program has promise and will be discussed further in section 3.

2.5 EnergyPlus

The Building Systems Laboratory (University of Illinois, 2003) together with Lawrence Berkeley National Laboratory and the United States Department of Energy (2003) has combined two programs, BLAST and DOE-2, to achieve one program, EnergyPlus, that is available free of charge from the internet. EnergyPlus (United Stated department of Energy, 2003, Crawley et al., 2001 and Olsena and Chen, 2003) combines the best features from each of its predecessors, as well as introducing some of its own features. With this program, the heating, cooling, lighting, ventilating and other energy related flows in a building can be simulated. It uses a heat balance-based zone simulation method to perform calculations. When analyzing buildings, EnergyPlus can account for moisture adsorption and desorption within the building elements, one of the desired characteristics. A major disadvantage of the program is that it was not designed to be the main graphical interface used, but should be used with an interface from a third-party developer. This makes the program more difficult to use.   The contact for the BLAST program said that

Energy Plus would be able to perform all the desired functions, from energy recovery to moisture storage and will therefore be considered in more detail in section 3.

2.6 HAP

Hourly Analysis Program (HAP) from Carrier (2003), is a load estimating simulation tool. It provides results of both building loads and equipment operation for commercial buildings. All loads calculations are performed using the Transfer Function load method. The input required for the program includes properties of all structural surfaces, as well as internal gains. Internal gains can include occupancy, equipment and miscellaneous sensible and latent loads. Unfortunately, the user of HAP does not have access to the source code for the program, making it somewhat inadequate for researchers. According to Carrier, HAP has the ability to analyze either sensible heat recovery alone, or the combination of sensible and latent heat recovery. The program, analyzes moisture flow due to air exchange in and out of the building, but it cannot analyze moisture flow through constructions, such as walls and roofs. This would be needed for whole building HAM simulation and thus HAP will be omitted from further deliberation.

2.7 IDA Indoor Climate and Energy

IDA Indoor Climate and Energy (IDA ICE) (Bjorsell et al., 2003 and Bring et al., 2003) was created by a Swedish simulation technology group called EQUA (2003). This building simulation program performs a heat balance on each zone in the building in order to simulate thermal comfort, IAQ and energy consumption. It can also simulate a number of other fundamental aspects of building design analysis. This is a highly trusted simulation package in Sweden, as it has been tested and verified against other highly acclaimed programs, which in turn, have been extensively tested against measurements and hand calculations.

Although the program is flexible and easy to use, it can have a long run time, depending on the complexity of the model structure. According to EQUA, the IDA ICE package uses a plate heat exchanger which can account for condensation but not moisture transfer. Other versions of IDA can use desiccant wheels, which allow moisture transfer, but these have not yet been modelled for ICE. There are additional models that can be incorporated with the ICE program that can simulate moisture storage in building materials. At present, a method of inputting moisture storage capabilities is being created, but not currently available. It looks like this program might in the future be able to perform all the tasks required, but according to the company, the current model (v3.0) cannot. In the future, this program should be analyzed further, but will not be presented in more detail in this report.

2.8 NewQUICK

NewQUICK (Mathews Consolidated Investments, 2003) is a building simulation tool that provides users with the sensible and latent loads and energy consumption of the building they are designing. It also has the ability to calculate air temperatures and the relative humidity of the space being conditioned. It has the advantage of being able to incorporate an unlimited number of zones within a building, but works on a single zone approach, meaning that there is no HAM transfer between different zones within a building. This tool provides a quick way of designing an efficient HVAC system. Although all this information was taken from the company website, there was no contact information for the company and the email links on the site failed. When the email would not go through, instructions came up to email the webmaster for the site, but this attempt also failed as it said the webmaster no longer existed. Since a connection could not be made with the company, no further information on the capabilities of the NewQUICK program could be obtained. This program will not be considered further.

2.9 RIUSKA

RIUSKA (Jokela et al., 2003 and Laine et al., 2003) is a building simulation tool that has been created by Granlund (2002) in collaboration with Lawrence Berkeley National Laboratory. The actual simulation engine behind RIUSKA is currently DOE 2.1E, while future versions will use DOE 2.2. RIUSKA is a thermal simulation program that can be used anywhere from the preliminary design of a building to the renovation of an old building. It is used to simulate heating and cooling loads, indoor comfort, and energy consumption of the building. The user interface of RIUSKA can vary based on the level of user knowledge and needs. Although an email was sent to Granlund a reply was not received. Therefore, the actual capabilities of RIUSKA remain unknown. Furthermore, the capabilities will be similar to DOE 2 since it is the simulation engine used in RIUSKA.

2.10 TRNSYS

The TRaNsient SYstems Simulation (TRNSYS) (Thermal Energy Systems Specialists, 2003, Florides et al., 2002 and Sodec 1999) program was developed by the Solar Energy Laboratory at the University of Wisconsin (2005), in 1975. It is a flexible simulation tool that can simulate the transient performance of thermal energy systems. TRNSYS can simulate a variety of energy systems with differing levels of complexity. The simulation program does not make any assumptions about the building or system being used so the user must have all the necessary details required for input into the program. This ensures that the simulation created represents the building being analyzed. According to the creators of the program, TRNSYS has the ability to calculate indoor temperature and relative humidity. As well, TRNYSY can account for the effects of moisture storage in building materials and rotating energy wheels. In addition, it permits the user to develop and run models of other building components and systems. This program seems to meet the requirements listed previously and therefore will be discussed further in section 3.

2.11 UMIDUS

UMIDUS (Home Energy, 2003, Mendes et al., 2001 and 2002) was developed at the Thermal Systems Laboratory of Pontifical Catholic University of Parana in Brazil. This program has the capability to analyze the hygrothermal performance of building elements. It takes into account one-dimensional heat and moisture transfer and has the ability to predict moisture and temperature profiles within different layers of the construction. The equations used to perform calculations are based on the conservation of energy and mass equations. This program is free to download from the internet. According to the company, UMIDUS has the ability to simulate moisture storage in the building envelope and materials, but is not able to predict the indoor relative humidity. They are however, working on a new program called Domus which will perform a hygrothermal analysis of the whole-building. This new program should have the moisture storage capabilities desired but it has not yet been released. This program should be considered in more detail in the future, but is not analyzed further in this report.

2.11 Summary

A summary of the capabilities of the 11 building energy tools is presented in Table 1. This table displays the results of inquiries made to each company as to whether or not their program had the desired capabilities. Where the information was not provided by the company and the information was found by another method, it is included in the table. An example e-mail sent by Melanie Schwab about DOE is: “I am researching a building simulation program for energy simulation and had a few questions about the capabilities of DOE 2.1. Does it include the simulation of air-to-air energy (both heat and moisture) recovery? Can it simulate the moisture storage in building envelopes and materials, similarly as thermal storage is included in simulation models? Would it be possible to input the moisture storage capabilities of the building materials and simulate the effect this has on the indoor relative humidity in the building?”

Table B.1. Summary of the capabilities of the reviewed building energy tools.

Program	Moisture storage in building materials	Calculate indoor humidity	Moisture exchanger in HVAC system	Access to Code
BLAST	No	No	No	No
BSim	**Yes**	**Yes**	**Yes**	**Yes**
CHVAC	No	No	No	No
DOE-2	No	No	No	**Yes**
EnergyPlus	**Yes**	**Yes**	**Yes**	**Yes**
HAP	No	—	**Yes**	No
IDA ICE	**Yes**	—	No	**Yes**
NewQUICK	—	**Yes**	—	—
RIUSKA	—	—	—	—
TRNSYS	**Yes**	**Yes**	**Yes**	**Yes**
UMIDUS	**Yes**	No	—	—

* A dash indicates that the company did not respond to that question.

3.0 Additional Information on 4 Programs

Table 1 indicates that BSim, EnergyPlus and TRNSYS may be the most applicable programs for this research. All three of these programs have the ability to simulate moisture storage, its effects on the indoor climate and air-to-air energy recovery. In addition, the user of these programs can access the source code. These three packages, along with DOE will therefore be considered further in this section.

3.1 DOE

Although there was no reply from the US Department of Energy, the DOE program is still considered further in this report because it is well known as a reliable and accurate program. More information was found about this program from personal communication with people who had used it in the past, as well as from the reference manual. Upon consulting the manual, it was determined that while new materials could be created in the program, it was not possible to specify any moisture properties for those materials. In fact, there was no mention of moisture capabilities in the reference manual at all. The reference manual used for this search, however, was from an older version of DOE (DOE 2.1-E). Even though the inclusion of moisture and humidity has become more important in building simulations it is unlikely that the newest version of DOE would meet the moisture requirements either.

3.2 EnergyPlus

The EnergyPlus program, also put forth by the US Department of Energy, meets all of the requirements stated earlier in this report and summarized in Table 1. Not only does it include air-to-air energy exchangers that model the transfer of both heat and moisture, it also has two different models for moisture storage. These two models are Moisture Transfer Functions, which simulates both storage and transport of moisture, and the Effective Mean Penetration Depth model, which simulates the moisture storage only (National Resources Canada, 2002). EnergyPlus allows the user access to the source code, as well as an extensive set of documentation on both the code, and the program in general. The source code, however, must be purchased.

3.3 BSim

The final two programs, BSim and TRNSYS both seemed suitable for Annex 41 research, so a demo version was obtained for each program. The BSim package contains SimView, which contains a 3-D representation of the building and SimDXF, which allows the user to create a building using CAD drawings (Danish Building and Urban Research, 2002). Within SimView it is possible to link to the following programs:

- Bv98 – heating demand calculation
- tsbi5 – performs thermal and moisture simulations

- XSun – analysis of solar distribution and shadows in buildings
- SimLight – estimation of the daylight conditions
- SimDB – database containing materials, constructions and surfaces

As well, within SimView, the user is able to:

- Couple tsbi5 results with a CFD program
- Export to Radiance for detailed calculation of lighting conditions

In the demo version of the BSim package, it is not possible to run Bv98 and there are some limitations within the SimDXF program. One such limitation is that a new drawing cannot be created; only existing models can be opened. For this reason, a good understanding of how SimDXF works was not obtained. It appears though, that it is similar to other CAD drawing programs, such as AutoCAD.

The first step to running a simulation is to create the building. This can be done by importing a drawing from SimDXF, or by creating the building right in SimView. When a building is created in SimView, its geometry is clearly displayed in four windows, three showing front, side and top views and one showing the complete 3-D view. This makes creating the building simple and enables the user to instantly see any problems with the building or its features.

Once the building has been created and any windows or doors have been added, the material construction must be attached to each surface of the model. This is done by calling the database, SimDB, and choosing the appropriate material layers for each surface. These can then be applied to the building, and the results will be shown in the display windows. SimView has a standard material database that contains a number of different construction layers, as well as a moisture database that contains constructions with moisture properties. The database is selected and attached to the model when the program is started.

Once the building is finished, thermal zones can be added to the model. Each space can be placed into a thermal zone and the systems within that zone can be defined. These systems include loads from people, lighting, equipment, infiltration, venting, ventilation, heating, cooling, mixing, and additional moisture sources. After the systems have been defined a check can be performed on the whole model to find any errors that might prevent the simulation from working properly. This was done for sample simulations and the ease of understanding the results varied. Sometimes the errors were very clear and could easily be changed, e.g. once the material for the frames around the windows was left undefined. Other times, the errors were less obvious and the method of fixing them was unclear.

Following this, the tsbi5 button is selected and the simulation options are chosen. This includes options such as longwave radiation and XSun distribution. If the moisture database has been attached to the model more options are offered including moisture transport and latent heat calculations. The length of the simulation can also be set, but for the demo version, this was limited to 14 days. The simulation is then started and the results can be seen in both tabular and graphical format for any day out of the simulation period. Results were obtained for some example buildings that were created with the help of the BSim Manual.

3.4 TRNSYS

The TRNSYS demo package also came with several other programs, including: SimCAD, a building drawing package; PREBID, used to specify and describe each thermal zone within the building; IISiBat, a simulation program used to assemble the components of the HVAC system and create TRNSYS input files; and TRNSYS, to run the building simulations. Since this was a demo version of the software, there were several limitations in the different programs. These limitations prevented a trial simulation from actually being preformed, although the data file output of a few examples could be viewed.

The SimCAD program came with an extensive help manual, which contained several example buildings to illustrate how the program works. These buildings were created to get an understanding of the program, which turned out to be quite easy to use. Walls, floors, windows and roofs could be defined for the building and more storeys could be added on to it. The building could be viewed in either 2-D, or 3-D. When the walls were being created it was possible to attach a certain material to them, as well as that material's thickness. Unfortunately, in the demo version, the buildings could not be saved, so it was not possible to run a simulation from a user created building. In the full version, once a building is saved, a file is created for input into the PREBID program.

In the PREBID program, an existing building can be inserted from SimCAD, or a new building can be created. The geometry of the building, including construction materials for the walls and windows can be specified. Once this is complete, the building's thermal zones are created. Here such information as internal gains, infiltration, and ventilation, indoor comfort and heating and cooling can be added to each zone. Indoor comfort can be specified by choosing the clothing factor and metabolic rate of the people within the room, as well as the relative air speed in the room. Additionally, one of two humidity models may be chosen for the simulation. These models include a capacitance humidity model, where the capacitance ratio is given, and a buffer storage model, where the gradient of sorptive isothermal line, mass and exchange coefficient can be specified for both surface and deep buffer storage. Once all the information for each zone has been recorded, the building is ready to be input into the IISiBat program.

IISiBat is used to layout the HVAC system used in the building. This is done by placing components on the worksheet and then connecting them so they can share information, as shown in Figure 1. The output of one component becomes the input of the next component in the system. In the schematic below, the air is removed from the building and moved through a fan and a heating coil, before re-entering the building.

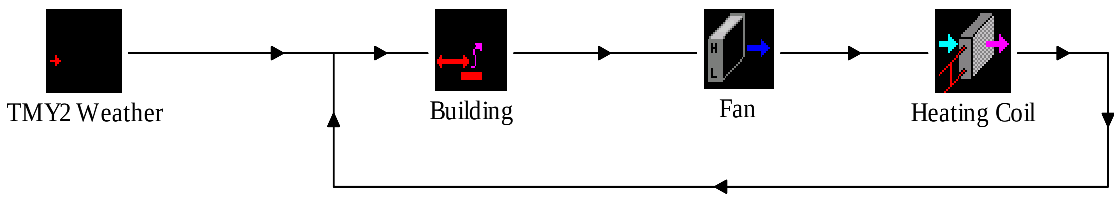

Figure B.1. IISiBat schematic of a building with heated recirculation air.

IISiBat comes with a standard component library which contains many components commonly found in energy systems for buildings. Additional libraries are available for purchase, which contain more components that can be used in the system. One such library is an HVAC library which contains a humidifier, and an air-to-air heat recovery system, as well as many other useful components.

The space to be conditioned in the simulation can be placed in the schematic in one of several methods. First, a multi-zone component can be chosen and the component can be linked to the PREBID file. This component will then have all the information that was given to the building in the PREBID program. The second method is a single-zone component which can be created by connecting walls, windows, floors and a roof right in the IISiBat worksheet. Finally, a lumped-capacitance building model can be chosen from the IISiBat library. Since this was a demo version, the TRNSYS executable file was disabled so a simulation could not be run.

The full version TRNSYS package comes complete with PREBID, IISiBat, FORTRAN source code for all components, and extensive documentation, in addition to the TRNSYS program. Available libraries are: HVAC component library, utility component library, storage component library, solar component library, geothermal heat pump component library, and applications component library.

4.0 Conclusions

In this report, over 260 building energy simulation tools were evaluated to consider their ability to simulate whole building HAM transfer. Non of the programs could include all aspects of HAM transfer, but 11 programs seemed to have the potential to perform moisture calculations together with thermal calculations and specifically had the potential to model moisture storage in hygroscopic materials for indoor air and HVAC systems (i.e., energy exchangers) and their effect on energy use. The companies for each of these 11 programs were then contacted and of those that responded, four programs (DOE, EnergyPlus, BSim and TRNSYS) appeared to be quite suitable and were investigated further.

APPENDIX C

DETAILED DESCRIPTION OF THE COMPUTER PACKAGE

DETAILED DESCRIPTION OF THE COMPUTER PACKAGE

The TRNSYS computer program is actually a package of several programs that work together to create models of buildings and HVAC systems and simulate their performance. TRNSYS is a FORTRAN based program and is not overly easy to use. For this reason it comes with a program called IISiBat (The Intelligent Interface for the Simulation of Buildings) which is a user friendly interface for creating a model of the HVAC system. Before a model of the HVAC system can be created in IISiBat though, the building must first be created. This is done using the SimCAD and PREBID programs. The SimCAD program is used to create the geometry of the building and to specify the different zones within the building. This building is then input into the PREBID program where all the internal details are specified, including wall and window construction, ventilation, infiltration and occupancy and equipment schedules. Once the building is finished it is then input as a component in the IISiBat workspace.

SimCAD v 1.3

The first step in the building model process is to create the geometry of the building. This is made easy with the SimCAD program. Similar to other CAD programs, such as AutoCAD, the user creates the walls of the buildings simply by connecting 2-D lines in the workspace. In this way the user can create the external walls of the building and any internal walls separating different rooms. It is also possible to create a building in SimCAD using a drawing from another CAD program.

In order to create a 3-D view of the building a room name must be specified for each room. Once each room has a name it is possible to specify thermal zones for the building. A thermal zone can be created for each room separately, or several rooms can be added to one thermal zone. For example in the school building, each of the classrooms is included in one thermal zone called “classroom”, but each of the other rooms (kitchen, cafeteria, etc.) are specified as their own thermal zone. This means that the temperature and relative humidity of the classrooms will be controlled as a whole and not individually for each room. The 3D model of the building can now be created.

Once the 3D model has been created successfully, windows and other openings can be added to the building. When creating the windows it is possible to select the type of window, as well as the dimensions of the window and the distance from the floor to the bottom of the window. If the opening is to be a door then the distance from the floor to the bottom of the opening would be set at zero.

Once the first floor is complete additional stories can be added. To complete the building the roof must be specified. The roof can be very simple, just a flat surface, or it can be more complex with sloped surfaces and gables. The building is now finished and can be saved, which creates the appropriate files to be used in PREBID.

PREBID v 5.0.8

After the geometry of the building is created, the properties of the building need to be specified, including wall construction, occupancy and equipment levels. As well, the ventilation, infiltration, and the heating and cooling systems can be specified now, or later in IISiBat. This information is put into the TRNSYS program via two files: a building description file (*bld) and a file containing the ASHRAE transfer function for the walls (*trn). The PREBID program makes the production of these files easy. PREBID was created by TRANSSOLAR Energietechnik GmbH, the German distributor of TRNSYS.

To begin working with the PREBID program, the user can either start a new project or use the pre-existing file created in SimCAD. In a new project, the user can specify each thermal zone and then the walls and windows within each zone. Since this project began with the creation of the building in SimCAD there was no need to create the zones. By opening the file created in SimCAD, all the information regarding the thermal zones and the location and size of all walls and windows is entered into the PREBID program. In order to complete the building in PREBID, the user must specify information about each zone individually. There are three main areas that need to be specified, the Regime Data, the Walls and the Windows.

The first section is the Regime Data. This includes the zone volume, thermal capacitance, initial temperature, initial relative humidity and the humidity model used. The zone volume is simply the volume of the air within the zone. This is automatically calculated if a SimCAD file is used. The thermal capacitance includes the capacitance of the zone air, as well as any mass not considered as walls (i.e., furniture). The default value is set to 1.2*zone volume. The thermal capacitance was increased to 12 times the zone volume for all the spaces containing furniture, such as desks and tables. The default values for the initial temperature and relative humidity are set at 20°*C* (68°*F*) and 50*% RH*.

There are two options for the humidity model. The first is the simple humidity model where the humidity capacitance ratio is set. The second is the moisture capacitance model, or the buffer storage model. In the buffer storage model, the gradient of the sorptive isothermal line, the mass and the exchange coefficient are specified for both surface buffer storage and deep buffer storage. The simple humidity model with a humidity capacitance ratio of 1 was chosen in this thesis.

The next section is the walls. In order to specify the walls three pieces of information are needed: the construction of the wall, surface area, and the category. The wall construction can be taken from a pre-existing library, or can be defined by the user. The surface area and category are automatically entered for each wall if a SimCAD file is used. The category refers to the position of the wall, whether it is an external wall, an internal wall (within a zone), a wall adjacent to another zone or a wall with specific boundary conditions.

To define a window, the window type, surface area, category, U-value and g-value need to be specified. The u-value is the coefficient for thermal losses through the window and the g-value is the solar heat gain coefficient. The window type can be selected from a library, or can be defined by the user. The library consists of five choices: single pane, double pane, triple pane, double pane with low-e glazing and double pane with low-e glazing and argon in the gap instead of air. The office building has double pane windows while the school building has triple pane windows. If a window is chosen from the library, the u-value and g-value are automatically entered. The surface area is automatically entered from the SimCAD file and the window category is the same as the wall category for the associated wall.

The infiltration rate into the zone is specified in terms of air changes per hour. For these simulations the infiltration was set as a schedule for the lobby space in the office building and as a constant for all the other spaces in the office building and in the school. The ventilation rate is also specified by the air changes per hour. As well, the temperature and relative humidity of the air flow must be specified. All of these values are specified as inputs to the building from the IISiBat workspace.

Internal heating and cooling can be added to the building in PREBID. For heating, the set point temperature for room control, the amount of heating power and the addition of humidification can be specified. For cooling, the set point temperature, amount of cooling power and whether or not dehumidification is required are specified. These functions are utilized in the simulations to determine the size of the heating and cooling equipment that is required to meet the heating and cooling loads of the building.

Internal heat gains can be specified as people, computers, artificial lighting or other gains. The heat gain from occupants is based on the number of occupants and the activity that they are doing. In the PREBID program, the activity that the people are doing is chosen from a table based on ISO 7730 (1994). These tables allow the user to select the degree of activity, whether they are seated at rest, doing heavy machine work, or something in between. Also listed is the total heat gain, as well as the sensible and latent portions of the heat gain.

To specify heat gains from a computer, the type of computer must be chosen. This is done through a drop down menu which gives the options: printer, terminal, PC with monitor and PC with color monitor. The computers chosen for these simulations are the PC with color monitor as described earlier. The heat gain from artificial lighting is also chosen from a drop down menu. Here the total heat gain [W/m^2] and the convective portion [%] of the heat gain are chosen from the list. These are chosen to be 19 W/m^2 and

40%, respectively. Other heat gains can be added by specifying the radiative power [kJ/hr], the convective power [kJ/hr] and the rate of moisture production [kg/hr].

The last detail to be created is the comfort levels in each space. This is done by specifying the clothing factor, metabolic rate and external work of the people within the room. As well, the relative velocity of the air within the space is required. The comfort levels in each zone are used to calculate the percentage of people that are dissatisfied with the zone conditions.

Before beginning work in IISiBat, the outputs that are going to be required by the IISiBat simulation must be set up in PREBID. There are 85 possible outputs which fall into three categories: Zone outputs, group of zone outputs and surface outputs. For each output selected, the thermal zone for which the output should be printed must be selected. If a surface output is chosen the individual surface for which the output is required must be specified. Once the outputs for each particular zone have been added to the list of outputs the building is then ready to be input into the IISiBat environment.

IISiBat v 3.0

IISiBat is a simulation program that has been adapted to house the TRNSYS software. It was created by the Building Technical and Scientifique Centre in France. Its user friendly environment makes it easy to create simulations by adding components to the workspace and then linking them together to pass information between them. Figure C.1 shows an example simulation created in IISiBat. This example mixes outdoor air with exhaust air at the mixer and then heats the air according to the thermostat, if necessary, before supplying it to the office building.

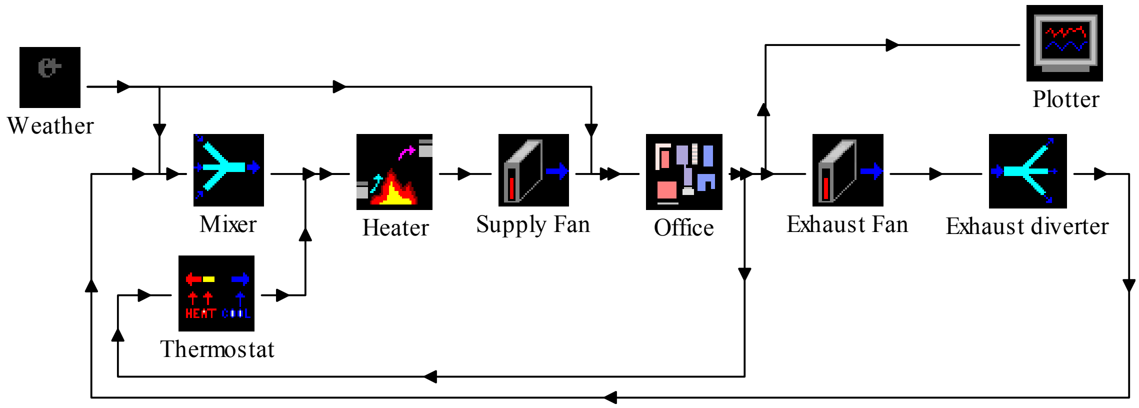

Figure C.1. Example schematic of a building and HVAC system in IISiBat.

The above example starts with a weather file. This weather file was downloaded from the United States Department of Energy website (2006). This website contains weather files for all major cities in the United States and Canada. The weather file used in IISiBat can come be in several different formats, including EnergyPlus, TMY2 (Typical Meteorological Year) and TRNSYS TMY. The above example uses the EnergyPlus format since that is the format found on the website.

The next component is the mixer. This component is a mixing valve for air, with up to 100 inlets. Here it is mixing the outdoor air with the exhaust air from the building. The user can specify the parameters, inputs, outputs and external files associated with each component. For the mixer, the only parameter needed is the number of inlets that are connected to it. By double clicking on the connection between two components, the user can specify the information that is being fed from one component to the next.

The office building is the component that was created in PREBID. As well, any inputs and outputs that were created in PREBID will be added to the list of inputs and outputs of the building component. The plotter creates a graph of selected information, such as temperature and relative humidity of a chosen space. Another form of outputting information is a printer, which prints data to a text file.

Details of the Components Used in the Simulations

The first set of components that is used in the simulations is the weather components. This includes the weather reader, which takes information from the weather file, the radiation processors and the fictive sky temperature calculator, seen in Figure C.2. The lines connecting the components represent information being passed from one component to another. The arrows show the direction of the information transfer. The particular weather reader used in this thesis is an EnergyPlus data reader. The weather files for each of the four cities were obtained in the EnergyPlus format (TMY2) from the United States Department of Energy website (2006). The weather data is given for each hour throughout the year. When the simulation is run at intervals of less than one hour the data in the weather file is interpolated to give data at each interval. This component reads the information from the weather file and relays that information on to other components in the simulation. The radiation processors take information about the radiation from the weather file and determine the amount of radiation on each surface of the building. The fictive sky temperature is required by the building for calculations of long wave radiation.

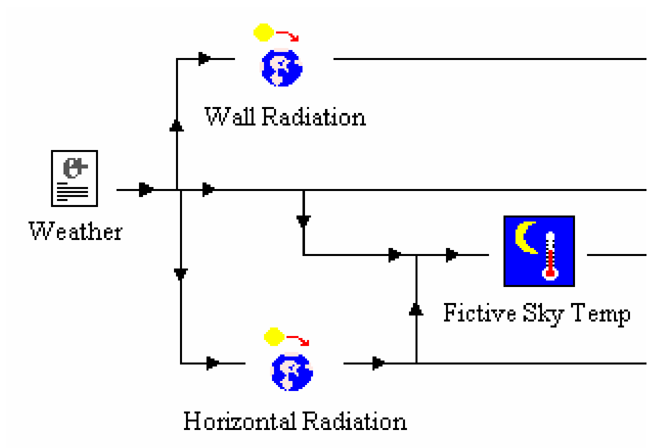

Figure C.2. Example weather information components in IISiBat.

In order to control the temperature in the space, a thermostat is required. The thermostat, as described in section 2.3.1 is set at 22°*C* for heating and 24°*C* for cooling. As well, there is a heating setback that reduces the control temperature to 18°*C* during the night. This is accomplished using a forcing function shown by the Heating Setback component in Figure C.3. The forcing function outputs a value of 1 during the night and 0 during the day. This is linked to the night setback control input of the thermostat and is multiplied by the setback temperature difference, so that during the night the setback difference is 4°*C* and during the day it is 0°*C*.

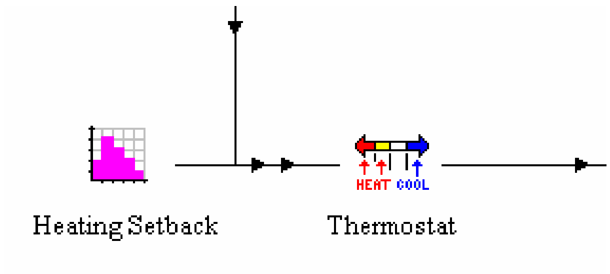

Figure C.3. Example temperature control components in IISiBat.

The effectiveness correlations, economizer and bypass are created using equation components. In these components the user creates the inputs and outputs and then the equations that are used to calculate the outputs, based on the inputs. These components are created using the equations from section 2.3.1. The equation components are also used to create a schedule for the ventilation. The program would normally supply air at all times, so a forcing function is used to shut the ventilation off during the night hours. When the simulation was run for the first time however, it was found that the temperature in the space got too cold in Saskatoon (below 0°*C* (32°*F*)) and too hot in Tampa (the building could not be cooled again the next day) during the night hours. For this reason another equation component was implemented to allow air to be supplied to the space at night if heating or cooling is required. During these times though, no outdoor air is used, only 100% recirculation air as there are no occupants in the space.

The mixing component, Figure C.4 is used for mixing two or more air flows. This component is used for mixing the outdoor air with the recirculation air. A mixing component is also used for the air exiting the building to mix the air coming from all the spaces into one exhaust stream. The component uses mass and energy balances to calculate the outlet conditions.

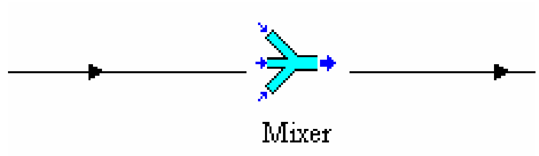

Figure C.4. Example mixing component in IISiBat.

The cooling unit used is a Type651 cooling coil (Figure C.5). This component requires two files that must be connected to it: one containing the total capacity and power as a function of outdoor dry bulb temperature, indoor wet bulb temperature, and the evaporator flow rate; and one containing the ratio of sensible to total cooling capacity as a function of the indoor dry bulb and wet bulb temperatures and the evaporator flow rate. The simulation program came with sample files which had to be scaled to fit the requirements of this simulation. The cooling coil is given a coefficient of performance (COP) of 2.7 based on ASHRAE Standard 90.1 (2004). The COP is the ratio of the energy removed from the air by the cooling coil to the power consumed by the cooling coil.

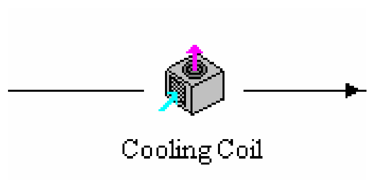

Figure C.5. Example cooling equipment in IISiBat.

The heating equipment used is a Type643 auxiliary heater, Figure C.6. This component adds heat to the incoming air stream at a user specified rate. A set point temperature is specified which the output temperature of the air stream will never exceed. The maximum possible heat transferred to the air is the user specified rate multiplied by the efficiency of the heater. This heater uses natural gas and has an efficiency of 79%. The temperature of the outlet stream is calculated based on an energy balance. No humidification or dehumidification occurs in this model.

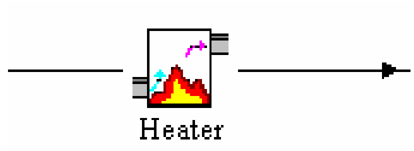

Figure C.6. Example heating equipment in IISiBat.

The fan component chosen is a variable speed fan, as seen in Figure C.7, but for the purposes of this thesis it is used as a constant volume fan. This component sets the flow rate for the rest of the components in the simulation. This is done by multiplying the user specified flow rate by the control signal (0 or 1) for this component. The rated air flow rate is calculated from the ventilation requirements of the buildings. The total required outdoor flow rate for each building is determined and then the total flow rate can be found, based on 20% outdoor air. The rated fan power must also be specified for this component. The fan power is calculated based on the flow rate of air through the fan, the pressure drop across the fan, and the efficiency of the fan. There are two fans in the system, a supply fan, before the building and an exhaust fan, after the building. The supply fan has a pressure drop of 1250*Pa* (5inH_2O) and the exhaust fan has a pressure drop of 500*Pa* (2inH_2O). Both fans have an efficiency of 70%. These values are based on the simulations run by Dhital. The total flow rate for the office building is 80m^3/s (169,560*cfm*) and for the school building it is 100m^3/s (211,950*cfm*).

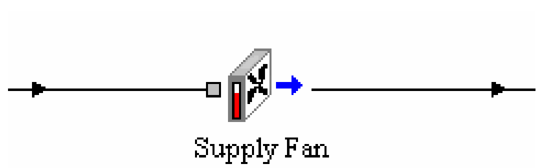

Figure C.7. Example supply fan component in IISiBat.

When the air is supplied to the building it must be split into the different spaces. This is accomplished with a diverting component, seen in Figure C.8. The fraction of air that goes to each branch is specified in the component. This is based on the fraction of the total outdoor air that each space requires to meet ventilation needs. The outlet temperature and humidity conditions for the air leaving this component are the same as the inlet conditions. There is also a diverter used for the air extracted from the building. A fraction of the air (X) is recirculated and the remainder is exhausted to the outdoors.

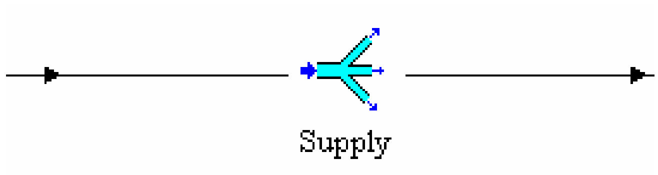

Figure C.8. Example diverting component in IISiBat.

The final major component in the base case simulations is the building. This component, seen in Figure C.9, uses the files created by the PREBID program where the building details are specified. The inputs and outputs are also specified in the PREBID program. To determine the size of the heating and cooling equipment needed by the building, an internal simulation is run on the building. This is a simulation that has internal heating and cooling controls, instead of the heating and cooling components listed above. All other components are the same. This simulation calculates the internal loads which can then be used to size the heating and cooling equipment in the main simulations. To determine the cooling load the building is kept the same as this will give the worst case scenario for the cooling load. For the heating load, all of the internal loads are removed from the building, as is done in the ASHRAE design methods (ASHRAE, 2005) to get the worst case scenario or peak heating load of the building. The cooling load value is the used to scale the files required by the cooling coil component and the heating load value is input into the heating component as the maximum rate of heat transfer.

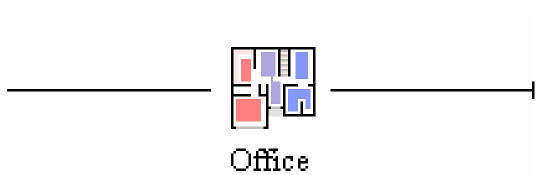

Figure C.9. Example building component in IISiBat.

The energy wheel component used in the final two simulations is a Type667 and is seen in Figure C.10. This component models air-to-air energy recovery based on the effectiveness method. The sensible and latent effectivenesses are required inputs to this component.

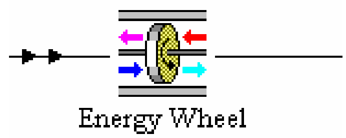

Figure C.10. Example energy wheel component in IISiBat.

The governing equations for this component are:

$$Q_s = \varepsilon_s \dot{m}_{min} Cp_{air} \Delta T \tag{C.1}$$

and

$$Q_l = \varepsilon_l \dot{m}_{min} h_{fg} \Delta W, \tag{C.2}$$

where: Q_s = sensible heat transfer [W]
Q_l = latent heat transfer [W]
ε_s = heat transfer effectiveness for sensible heat transfer
ε_l = heat transfer effectiveness for latent heat transfer
Cp_{air} = constant-pressure specific heat of air [J/(kg·K)]
h_{fg} = latent heat of vaporization of air [J/kg]
$\dot{m}_{min}$ = minimum flow rate of air of the two streams [kg/s]
T = air temperature [K or °C]
W = air humidity ratio [kg_w/kg_a].

APPENDIX D

SAMPLE IISIBAT SIMULATION FILE FOR THE OFFICE BUILDING

SAMPLE IISIBAT SIMULATION FILE FOR THE OFFICE BUILDING

```
1               TRNSYS - A TRANSIENT SIMULATION PROGRAM
        FROM THE SOLAR ENERGY LAB AT THE UNIVERSITY OF WISCONSIN
                       VERSION  15  Feb. 2000
 ASSIGN Office_s.LST                                        6
*** Control cards
* START, STOP and STEP
 CONSTANTS   3
   START=8737
   STOP=17520
   STEP=.04
* User defined CONSTANTS
*SIMULATION Start time            End time            Time step
 SIMULATION        8.737E+03    1.752E+04    4.000E-02
*                                          Integration           Convergence
 TOLERANCES        1.000E-03    1.000E-01
*                                          Max iterations       Max warnings      Trace limit
 LIMITS     50    100    101
*                                          TRNSYS numerical integration solver method
 DFQ   1
*                                          TRNSYS output file width, number of characters
 WIDTH    80
*                                          NOLIST statement
 LIST
*                                          MAP statement
 MAP
*                                          Solver statement
 SOLVER  0
*** Units
* Model "Weather" (Type 89)
 UNIT   1     TYPE  89     Weather
*$UNIT_NAME Weather
*$MODEL .\Utility\Weather and Other Data Readers\Standard Weather File Types\EnergyPlus\Skip N
Lines before starting\TYPE89f.tmf
*$POSITION 41 -5
*$LAYER Main
*$# This component conforms to the format of the TRNSYS TMY and the TMY data file reader. It is
designed
*$# to read EnergyPlus (*.epw) format weather files.
     PARAMETERS   2
* 1 Mode
* 2 Logical unit
    -3.000E+00    1.300E+01
*** External files
 ASSIGN C:\Thesiswork\Buildings\CV\Offices\Saskatoon\Weather_saskato   13
*|? Which file contains the EnergyPlus weather information? |1000
* Model "Outdoor" (Type 33)
 UNIT   2     TYPE  33     Outdoor
*$UNIT_NAME Outdoor
*$MODEL .\Physical Phenomena\Thermodynamic Properties\Psychrometrics\Dry Bulb and Dew Point
Known\TYPE33d.tmf
*$POSITION 140 19
*$LAYER Main
     PARAMETERS   4
* 1 Psychrometrics mode
```

```
 * 2 Pressure
 * 3 Wet bulb mode
 * 4 Error mode
      3.000E+00    1.000E+00    0.000E+00    1.000E+00
      INPUTS    2
 * Weather:Dry bulb temperature ->Dry bulb temp.
 * Weather:Dew point temperature ->Dew point temp.
      1,5           1,9
 *** INITIAL INPUT VALUES
      2.200E+01    2.000E+01
 * Model "Wall Radiation" (Type 16)
  UNIT   3     TYPE  16     Wall Radiation
 *$UNIT_NAME Wall Radiation
 *$MODEL .\Physical Phenomena\Radiation Processors\Total Horiz, Direct Normal Known
(Mode=4)\No Radiation Smoothing\TYPE16g.tmf
 *$POSITION 141 83
 *$LAYER Weather / Data Files
      PARAMETERS   9
 * 1 Horiz. radiation mode
 * 2 Tracking mode
 * 3 Tilted surface mode
 * 4 Starting day
 * 5 Latitude
 * 6 Solar constant
 * 7 Shift in solar time
 * 8 Not used
 * 9 Solar time?
      4.000E+00    1.000E+00    3.000E+00    3.650E+02    5.217E+01
      4.871E+03   -1.668E+01    2.000E+00    1.000E+00
      INPUTS   13
 * Weather:Global horizontal radiation ->Total radiation on horizontal surface
 * Weather:Direct normal radiation ->Direct normal beam radiation
 * Weather:Time of last read ->Time of last data read
 * Weather:Time of next read ->Time of next data read
 * [unconnected] Ground reflectance
 * [unconnected] Slope of surface-1
 * [unconnected] Azimuth of surface-1
 * [unconnected] Slope of surface-2
 * [unconnected] Azimuth of surface-2
 * [unconnected] Slope of surface-3
 * [unconnected] Azimuth of surface-3
 * [unconnected] Slope of surface-4
 * [unconnected] Azimuth of surface-4
      1,4           1,3           1,99          1,100         CONST
      CONST         CONST         CONST         CONST         CONST
      CONST         CONST         CONST
 *** INITIAL INPUT VALUES
      0.000E+00    0.000E+00    0.000E+00    1.000E+00    2.000E-01
      9.000E+01    1.800E+02    9.000E+01    2.700E+02    9.000E+01
      0.000E+00    9.000E+01    9.000E+01
 * Model "Horizontal Radiation" (Type 16)
  UNIT   4     TYPE  16     Horizontal Radiation
 *$UNIT_NAME Horizontal Radiation
 *$MODEL .\Physical Phenomena\Radiation Processors\Total Horiz, Direct Normal Known
(Mode=4)\No Radiation Smoothing\TYPE16g.tmf
 *$POSITION 145 147
```

```
*$LAYER Weather / Data Files
PARAMETERS   9
* 1 Horiz. radiation mode
* 2 Tracking mode
* 3 Tilted surface mode
* 4 Starting day
* 5 Latitude
* 6 Solar constant
* 7 Shift in solar time
* 8 Not used
* 9 Solar time?
4.000E+00    1.000E+00    3.000E+00    3.650E+02    5.217E+01
4.871E+03   -1.668E+01    2.000E+00    1.000E+00
INPUTS    7
* Weather:Global horizontal radiation ->Total radiation on horizontal surface
* Weather:Direct normal radiation ->Direct normal beam radiation
* Weather:Time of last read ->Time of last data read
* Weather:Time of next read ->Time of next data read
* [unconnected] Ground reflectance
* [unconnected] Slope of surface
* [unconnected] Azimuth of surface
1,04          1,03          1,99          1,100         CONST
CONST         CONST
*** INITIAL INPUT VALUES
0.000E+00    0.000E+00    0.000E+00    1.000E+00    2.000E-01
0.000E+00    0.000E+00
* Model "Fictive Sky Temp" (Type 69)
UNIT   5     TYPE  69     Fictive Sky Temp
*$UNIT_NAME Fictive Sky Temp
*$MODEL .\Physical Phenomena\Sky Temperature\calculate cloudiness factor\TYPE69b.tmf
*$POSITION 286 104
*$LAYER Main
PARAMETERS   2
* 1 mode for cloudiness factor
* 2 height over sea level
0.000E+00    5.040E+02
INPUTS    5
* Weather:Dry bulb temperature ->Ambient temperature
* Weather:Dew point temperature ->Dew point temperature at ambient conditions
* Horizontal Radiation:Beam radiation on horizontal ->Beam radiation on the horizontal
* Horizontal Radiation:Horizontal diffuse radiation ->Diffuse radiation on the horizontal
* [unconnected] not used with this mode
1,05          1,09          4,05          4,06          CONST
*** INITIAL INPUT VALUES
0.000E+00    2.000E+01    0.000E+00    0.000E+00    0.000E+00
* Model "Heating Setback" (Type 14)
UNIT   6     TYPE  14     Heating Setback
*$UNIT_NAME Heating Setback
*$MODEL .\Utility\Forcing Functions\General\TYPE14h.tmf
*$POSITION 50 531
*$LAYER Main
PARAMETERS  12
* 1 Initial value of time
* 2 Initial value of function
* 3 Time at point-1
* 4 Value at point -1
```

```
* 5 Time at point-2
* 6 Value at point -2
* 7 Time at point-3
* 8 Value at point -3
* 9 Time at point-4
* 10 Value at point -4
* 11 Time at point-5
* 12 Value at point -5
     0.000E+00    1.000E+00    6.000E+00    1.000E+00    6.000E+00
     0.000E+00    2.300E+01    0.000E+00    2.300E+01    1.000E+00
     2.400E+01    1.000E+00
* Model "Thermostat" (Type 8)
 UNIT   7     TYPE   8     Thermostat
*$UNIT_NAME Thermostat
*$MODEL .\Controllers\3-Stage Room Thermostat\w_ heating set back\TYPE8a.tmf
*$POSITION 175 531
*$LAYER Main
     PARAMETERS   7
* 1 Nb. of oscillations permitted
* 2 1st stage heating in 2nd stage?
* 3 Minimum primary source temperature
* 4 Temperature for cooling
* 5 1st stage heating temperature
* 6 2nd stage heating temperature
* 7 Heating set back temperature difference
     5.000E+00    1.000E+00    2.000E+01    2.400E+01    2.200E+01
     1.800E+01    4.000E+00
     INPUTS    3
* Office:    3- (air temperature of zone)  TAIR   3 ->Room temperature
* [unconnected] 1st stage source temperature
* Heating Setback:Instantaneous value of function over the timestep ->Set back control function
     35,03          CONST          6,02
*** INITIAL INPUT VALUES
     2.000E+01    3.000E+01    1.000E+00
* Model "Night" (Type 14)
 UNIT   8     TYPE  14     Night
*$UNIT_NAME Night
*$MODEL .\Utility\Forcing Functions\General\TYPE14h.tmf
*$POSITION 130 435
*$LAYER Main
     PARAMETERS  12
* 1 Initial value of time
* 2 Initial value of function
* 3 Time at point-1
* 4 Value at point -1
* 5 Time at point-2
* 6 Value at point -2
* 7 Time at point-3
* 8 Value at point -3
* 9 Time at point-4
* 10 Value at point -4
* 11 Time at point-5
* 12 Value at point -5
     0.000E+00    0.000E+00    6.000E+00    0.000E+00    6.000E+00
     1.000E+00    2.300E+01    1.000E+00    2.300E+01    0.000E+00
     2.400E+01    0.000E+00
```

```
 * EQUATIONS "Supply Control"
 EQUATIONS   1
   SupplyCont = or(eql([8,2],1),and(eql([8,2],0),or(eql([7,1],1),eql([7,3],1))))
*$UNIT_NAME Supply Control
*$LAYER Main
*$POSITION 286 431
 * EQUATIONS "Controls"
  EQUATIONS   2
   HCnew = [7,1]*SupplyCont
   CCnew = [7,3]*SupplyCont
*$UNIT_NAME Controls
*$LAYER Main
*$POSITION 381 463
 * Model "W-RH" (Type 33)
  UNIT  11     TYPE  33     W-RH
*$UNIT_NAME W-RH
*$MODEL .\Physical Phenomena\Thermodynamic Properties\Psychrometrics\Dry Bulb and Humidity
Ratio Known\TYPE33c.tmf
*$POSITION 451 531
*$LAYER Main
     PARAMETERS   4
* 1 Psychrometrics mode
* 2 Pressure
* 3 Wet bulb mode
* 4 Error mode
     4.000E+00    1.000E+00    0.000E+00    1.000E+00
     INPUTS    2
* Exhaust diverter:Temperature at outlet -2 ->Dry bulb temp.
* Exhaust diverter:Humidity ratio at outlet -2 ->Absolute humidity ratio
     38,04          38,05
*** INITIAL INPUT VALUES
     2.200E+01    6.000E-03
 * EQUATIONS "Parameters"
  EQUATIONS  12
   RHsupply = [2,6]/100
   RHexh = [11,6]/100
   Us = 0.5627*RHsupply-0.0062
   Ue = 0.5627*RHexh-0.0062
   dudphi = (abs(Us-Ue))/(max(0.000001,abs(RHsupply-RHexh)))
   Tave = (([2,7]+273.15)+([11,7]+273.15))/2
   RHave = (RHsupply+RHexh)/2
   Crmto = 2.735*dudphi**0.2*(exp(1482/Tave)/47.9-1.26*RHave**0.5)**4.66
   H1 = max(abs([2,7]-[11,7]),0.001)
   H2 = lt(([2,7]-[11,7]),0)*([2,7]-[11,7])
   H3 = H1+2*H2
   Hstar = 2500*([2,1]-[11,1])/H3
*$UNIT_NAME Parameters
*$LAYER Main
*$POSITION 115 473
 * EQUATIONS "Effectiveness"
  EQUATIONS   4
   el1 = 0.76717*(1-1/(0.54*(max(0.1,Crmto))**0.86))*(1-
1/(1.837*(max(0.1,Crmto))**0.54*(gt(Hstar,0)*Hstar+lt(Hstar,0)*Hstar+eql(Hstar,0)*.0001)))
   el2 = gt(el1,1)*1+and(gt(el1,0),lt(el1,1))*el1
   es1 = 0.7361273-(0.0191595)*Hstar
   es2 = gt(es1,1)*0.9999+and(gt(es1,0),lt(es1,1))*es1+eql(es1,0)*.01
```

```
*$UNIT_NAME Effectiveness
*$LAYER Main
*$POSITION 250 473
 * EQUATIONS "Temp Calc"
  EQUATIONS   2
    Tcritical = (14-(es2+0.8-es2*0.8)*[35,3])/((1-es2)*0.2)
    Tbypass = (14-0.8*[35,3])/0.2
*$UNIT_NAME Temp Calc
*$LAYER Main
*$POSITION 359 548
 * EQUATIONS "Ranges"
  EQUATIONS   5
    R1 = lt([2,7],Tcritical)
    R2 = and(lt([2,7],Tbypass),gt([2,7],Tcritical))
    R3 = and(lt([2,7],14),gt([2,7],Tbypass))
    R4 = and(lt([2,7],[35,3]),gt([2,7],14))
    R5 = gt([2,7],[35,3])
*$UNIT_NAME Ranges
*$LAYER Main
*$POSITION 458 612
 * EQUATIONS "Es/El/X"
  EQUATIONS  11
    e = or(eql(R1,1),eql(R5,1))*es2+((14-[2,7])-0.8*([35,3]-[2,7]))/((0.2)*max(([35,3]-[2,7]),0.1))*R2
    es = and(gt(e,0),eql(HCnew,1))*es2+eql(HCnew,0)*e
    ela = or(eql(R1,1),eql(R5,1))*el2+(el2*(Tbypass-[2,7]))/(max((Tbypass-Tcritical),0.1))*R2
    el = and(gt(ela,0),eql(HCnew,1))*el2+eql(HCnew,0)*ela
    Xa = or(or(eql(R1,1),eql(R2,1)),eql(R5,1))*0.8+(14-[2,7])/(max(([35,3]-[2,7]),0.1))*R3
    Xb = and(gt(Xa,0),eql(HCnew,1))*0.8+eql(HCnew,0)*Xa
    X = and(eql([8,2],0),eql(SupplyCont,1))*1+or(eql([8,2],1),and(eql([8,2],0),eql(SupplyCont,0)))*Xb
    moa = 342176.4*(1-X)
    mo = or(eql([8,2],1),and(eql([8,2],0),eql(CCnew,1)))*moa
    mout = or(or(eql(R1,1),eql(R2,1)),eql(R5,1))*mo
    mbypass = or(eql(R3,1),eql(R4,1))*mo
*$UNIT_NAME Es/El/X
*$LAYER Main
*$POSITION 564 516
 * EQUATIONS "EWcontrol"
  EQUATIONS   1
    EWcontrol = and(eql([8,2],1),or(or(eql(R1,1),eql(R2,1)),eql(R5,1)))
*$UNIT_NAME EWcontrol
*$LAYER Main
*$POSITION 653 346
 * Model "Energy Wheel" (Type 667)
  UNIT  18     TYPE 667     Energy Wheel
*$UNIT_NAME Energy Wheel
*$MODEL .\TESS Libraries\Hvac\Equipment\Air-Air Heat Recovery\Type667.tmf
*$POSITION 769 350
*$LAYER Main
     PARAMETERS   3
* 1 Specific heat of air
* 2 Rated power
* 3 Latent heat of vaporization of water
     1.007E+00    1.629E+04    2.454E+03
     INPUTS    9
* W-RH:Dry bulb temperature ->Exhaust air temperature
* W-RH:Percent relative humidity  ->Exhaust air %RH
```

```
 * Es/El/X:mout ->Exhaust air flow rate
 * Outdoor:Dry bulb temperature ->Ambient temperature
 * Outdoor:Percent relative humidity  ->Ambient %RH
 * Es/El/X:mout ->Outside air flow rate
 * Es/El/X:es ->Sensible effectiveness
 * Es/El/X:el ->Latent effectiveness
 * EWcontrol:EWcontrol ->On/Off Control Signal
      11,07          11,06          MOUT          2,07          2,06
      MOUT           ES             EL            EWCONTROL
 *** INITIAL INPUT VALUES
      2.000E+01    6.000E+01    6.844E+04    2.000E+01    5.000E+01
      2.737E+05    7.000E-01    7.000E-01    1.000E+00
  * EQUATIONS "bypass control"
   EQUATIONS   3
     mvent = [18,7]+mbypass
     Tvent = gt([18,7],0)*[18,5]+gt(mbypass,0)*[2,7]
     Wvent = gt([18,7],0)*[18,6]+gt(mbypass,0)*[2,1]
 *$UNIT_NAME bypass control
 *$LAYER Main
 *$POSITION 856 271
  * EQUATIONS "flow rate"
   EQUATIONS   1
     mmixed = mvent+[38,3]
 *$UNIT_NAME flow rate
 *$LAYER Main
 *$POSITION 957 356
  * Model "Mixer" (Type 648)
  UNIT  21     TYPE 648     Mixer
 *$UNIT_NAME Mixer
 *$MODEL .\TESS Libraries\Hvac\Hydronics\Valves\100-Port Flow Mixer\Air\Humidity
Ratio\Type648.tmf
 *$POSITION 1057 274
 *$LAYER Main
      PARAMETERS   1
 * 1 Number of inlets
      2.000E+00
      INPUTS    6
 * bypass control:Tvent ->Air temperature at inlet-1
 * bypass control:Wvent ->Air humidity ratio at inlet-1
 * bypass control:mvent ->Dry air flow rate at inlet-1
 * Exhaust diverter:Temperature at outlet -1 ->Air temperature at inlet-2
 * Exhaust diverter:Humidity ratio at outlet -1 ->Air humidity ratio at inlet-2
 * Exhaust diverter:Flow rate at outlet -1 ->Dry air flow rate at inlet-2
      TVENT          WVENT          MVENT          38,01          38,02
      38,03
 *** INITIAL INPUT VALUES
      2.000E+01    8.000E-03    6.844E+04    2.000E+01    8.000E-03
      2.737E+05
  * Model "Tdew" (Type 33)
   UNIT  22     TYPE  33     Tdew
 *$UNIT_NAME Tdew
 *$MODEL .\Physical Phenomena\Thermodynamic Properties\Psychrometrics\Dry Bulb and Humidity
Ratio Known\TYPE33c.tmf
 *$POSITION 1228 232
 *$LAYER Main
```

```
     PARAMETERS   4
* 1 Psychrometrics mode
* 2 Pressure
* 3 Wet bulb mode
* 4 Error mode
     4.000E+00    1.000E+00    1.000E+00    1.000E+00
     INPUTS    2
* Mixer:Air temperature at outlet  ->Dry bulb temp.
* Mixer:Air humidity ratio at outlet  ->Absolute humidity ratio
     21,01          21,02
*** INITIAL INPUT VALUES
     2.200E+01    6.000E-03
 * Model "Heater" (Type 643)
  UNIT  23     TYPE 643     Heater
*$UNIT_NAME Heater
*$MODEL .\TESS Libraries\Hvac\Equipment\Furnace\Type643.tmf
*$POSITION 1154 494
*$LAYER Main
     PARAMETERS   5
* 1 Maximum heating rate
* 2 Set point temperature
* 3 Specific heat of air
* 4 Overall loss coefficient
* 5 Efficiency of auxiliary heater
     1.098E+08    3.000E+01    1.007E+00    0.000E+00    7.900E-01
     INPUTS    5
* Mixer:Air temperature at outlet  ->Inlet air temperature
* Mixer:Air humidity ratio at outlet  ->Inlet air humidity ratio
* flow rate:mmixed ->Dry air flow rate
* Controls:HCnew ->Control function
* [unconnected] Environment temperature
     21,01          21,02          MMIXED         HCNEW         CONST
*** INITIAL INPUT VALUES
     2.000E+01    8.000E-03    3.422E+05    0.000E+00    2.000E+01
 * Model "Cooling Coil" (Type 651)
  UNIT  24     TYPE 651     Cooling Coil
*$UNIT_NAME Cooling Coil
*$MODEL .\TESS Libraries\Hvac\Equipment\Residential Cooling Coil\Type651.tmf
*$POSITION 1169 179
*$LAYER Main
     PARAMETERS  11
* 1 Logical unit for performance data
* 2 Logical unit for sensible ratio
* 3 Number of condenser temperatures
* 4 Number of evaporator flows
* 5 Number of indoor wet bulbs
* 6 Number of evaportator flow rates
* 7 Number of indoor wet bulbs
* 8 Number of indoor dry bulb temps.
* 9 Specific heat of air
* 10 Blower power draw
* 11 Controller power draw
     1.100E+01    1.200E+01    4.000E+00    3.000E+00    3.000E+00
     3.000E+00    3.000E+00    3.000E+00    1.007E+00    0.000E+00
     0.000E+00
     INPUTS    5
```

```
 * Outdoor:Dry bulb temperature ->Ambient temperature
 * Mixer:Air temperature at outlet  ->Indoor temperature
 * Mixer:Air humidity ratio at outlet  ->Indoor humidity ratio
 * flow rate:mmixed ->Evaporator flow rate
 * Controls:CCnew ->Control signal
      2,07           21,01          21,02          MMIXED         CCNEW
 *** INITIAL INPUT VALUES
      3.000E+01    2.000E+01    8.000E-03    3.422E+05    0.000E+00
 *** External files
  ASSIGN C:\Thesiswork\Buildings\CV\Offices\Saskatoon\Rescool_s_corr.   11
 *|? Which file contains the cooling coil performance data? |1000
  ASSIGN C:\Thesiswork\Buildings\CV\Offices\Saskatoon\Res_st.txt        12
 *|? Which file contains the sensible heat ratio data? |1000
  * EQUATIONS "Tcomp"
   EQUATIONS   2
     Wgt = gt([24,1],[22,8])*[24,2]
     RHlt = lt([24,1],[22,8])*100
 *$UNIT_NAME Tcomp
 *$LAYER Main
 *$POSITION 589 303
* Model "RH-W" (Type 33)
   UNIT  26     TYPE  33     RH-W
 *$UNIT_NAME RH-W
 *$MODEL .\Physical Phenomena\Thermodynamic Properties\Psychrometrics\Dry Bulb and Relative
Humidity Known\TYPE33e.tmf
 *$POSITION 759 264
 *$LAYER Main
      PARAMETERS   4
 * 1 Psychrometrics mode
 * 2 Pressure
 * 3 Wet bulb mode
 * 4 Error mode
      2.000E+00    1.000E+00    0.000E+00    1.000E+00
      INPUTS    2
 * Cooling Coil:Return air temperature ->Dry bulb temp.
 * Tcomp:RHlt ->Percent relative humidity
      24,01          RHLT
 *** INITIAL INPUT VALUES
      2.200E+01    6.000E+01
  * EQUATIONS "Wcool"
   EQUATIONS   1
     Wcool = max(Wgt,[26,1])
 *$UNIT_NAME Wcool
 *$LAYER Main
 *$POSITION 973 292
  * Model "hcoil" (Type 33)
   UNIT  28     TYPE  33     hcoil
 *$UNIT_NAME hcoil
 *$MODEL .\Physical Phenomena\Thermodynamic Properties\Psychrometrics\Dry Bulb and Humidity
Ratio Known\TYPE33c.tmf
 *$POSITION 1047 211
 *$LAYER Main
      PARAMETERS   4
 * 1 Psychrometrics mode
 * 2 Pressure
 * 3 Wet bulb mode
```

```
 * 4 Error mode
      4.000E+00    1.000E+00    0.000E+00    1.000E+00
      INPUTS    2
 * Cooling Coil:Return air temperature ->Dry bulb temp.
 * Wcool:Wcool ->Absolute humidity ratio
      24,01          WCOOL
 *** INITIAL INPUT VALUES
      2.200E+01    6.000E-03
  * EQUATIONS "qcoil"
   EQUATIONS   1
    qtotal = [24,3]*([22,3]-[28,3])*CCnew
 *$UNIT_NAME qcoil
 *$LAYER Main
 *$POSITION 1142 282
 * EQUATIONS "Air Properties"
   EQUATIONS   3
    Temp = eql(HCnew,1)*[23,1]+eql(CCnew,1)*[24,1]+and(lt(HCnew,1),lt(CCnew,1))*[21,1]
    Humidity = eql(HCnew,1)*[23,2]+eql(CCnew,1)*Wcool+and(lt(HCnew,1),lt(CCnew,1))*[21,2]
    Flowrate = mmixed
 *$UNIT_NAME Air Properties
 *$LAYER Main
 *$POSITION 1451 303
 * Model "Supply Fan" (Type 662)
  UNIT  31     TYPE 662     Supply Fan
 *$UNIT_NAME Supply Fan
 *$MODEL .\TESS Libraries\Hvac\Hydronics\Fans\Variable Speed Fan\Type662.tmf
 *$POSITION 1550 306
 *$LAYER Main
 *$# NOTE: This component sets the flow rate for the rest of the components in the flow loop by
multiplying the
 *$# nominal flow rate (Parameter 1) by the  control signal (Input 3).  The mass flow rate input of this
component
 *$# is only for visualization purposes; it is not used except for convergence checking.
      PARAMETERS   6
 * 1 Rated air flow rate
 * 2 Specific heat of air
 * 3 Rated fan power
 * 4 Thermal conversion coefficient
 * 5 Number of power coefficients
 * 6 Power coefficient
      3.422E+05    1.007E+00    5.092E+05    3.000E-01    1.000E+00
      1.000E+00
      INPUTS    4
 * Air Properties:Temp ->Inlet air temperature
 * Air Properties:Humidity ->Inlet air humidity ratio
 * Air Properties:Flowrate ->Inlet mass flow rate
 * Supply Control:SupplyCont ->Control signal
      TEMP          HUMIDITY       FLOWRATE       SUPPLYCONT
 *** INITIAL INPUT VALUES
      2.000E+01    8.000E-03    3.422E+05    1.000E+00
  * Model "Supply" (Type 646)
   UNIT  32     TYPE 646     Supply
 *$UNIT_NAME Supply
 *$MODEL .\TESS Libraries\Hvac\Hydronics\Valves\100-Port Flow Diverter\Air\Humidity
Ratio\Type646.tmf
 *$POSITION 1636 307
```

```
*$LAYER Main
     PARAMETERS   1
* 1 Number of outlet ports
     5.000E+00
     INPUTS   8
* Supply Fan:Outlet air temperature ->Inlet air temperature
* Supply Fan:Outlet air humidity ratio ->Inlet air humidity ratio
* Supply Fan:Outlet flow rate ->Inlet air flow rate
* [unconnected] Fraction of flow to outlet -1
* [unconnected] Fraction of flow to outlet -2
* [unconnected] Fraction of flow to outlet -3
* [unconnected] Fraction of flow to outlet -4
* [unconnected] Fraction of flow to outlet -5
     31,01        31,02        31,03        CONST        CONST
     CONST        CONST        CONST
*** INITIAL INPUT VALUES
     2.000E+01   8.000E-03   2.000E+03   3.300E-02   1.000E-02
     9.510E-01   4.000E-03   2.000E-03
 * Model "Density" (Type 33)
  UNIT  33    TYPE  33    Density
*$UNIT_NAME Density
*$MODEL .\Physical Phenomena\Thermodynamic Properties\Psychrometrics\Dry Bulb and Humidity
Ratio Known\TYPE33c.tmf
*$POSITION 1712 307
*$LAYER Main
     PARAMETERS   4
* 1 Psychrometrics mode
* 2 Pressure
* 3 Wet bulb mode
* 4 Error mode
     4.000E+00   1.000E+00   0.000E+00   1.000E+00
     INPUTS   2
* Supply:Temperature at outlet -1 ->Dry bulb temp.
* Supply:Humidity ratio at outlet -1 ->Absolute humidity ratio
     32,01        32,02
*** INITIAL INPUT VALUES
     2.200E+01   6.000E-03
* EQUATIONS "Supply Flowrate"
  EQUATIONS   6
   MD = MAX(0.1,[33,4])
   Qlobby = [32,3]/(MD*1895.4)*SupplyCont
   Qwash = [32,6]/(MD*469.8)*SupplyCont
   Qoffice = [32,9]/(MD*58745.2)*SupplyCont
   Qelev = [32,12]/(MD*694.12)*SupplyCont
   Qstairs = [32,15]/(MD*186.3)*SupplyCont
*$UNIT_NAME Supply Flowrate
*$LAYER Main
*$POSITION 1813 367
 * Model "Office" (Type 56)
  UNIT  35    TYPE  56    Office
*$UNIT_NAME Office
*$MODEL .\Loads and Structures\Multi-Zone Building\type56.tmf
*$POSITION 1890 243
*$LAYER Main
     PARAMETERS   5
* 1 Logical unit for description file
```

```
 * 2 Logical unit for transfer function file
 * 3 Logical unit for window library
 * 4 Star network calculation switch
 * 5 Weighting factor for operative temperature
      2.600E+01    2.700E+01    2.800E+01    0.000E+00    5.000E-01
      INPUTS   25
 * Outdoor:Dry bulb temperature ->    1- TAMB  (AMBIENT TEMPERATURE)
 * Outdoor:Percent relative humidity  ->    2- ARELHUM  (RELATIVE AMBIENT HUMIDITY)
 * Fictive Sky Temp:Fictive sky temperature ->    3- TSKY  (FIKTIVE SKY TEMPERATURE)
 * Wall Radiation:Total radiation on surface 1 ->    4- ITNORTH  (INCIDENT RADIATION FOR
ORIENTATION NORTH)
 * Wall Radiation:Total radiation on surface 3 ->    5- ITSOUTH  (INCIDENT RADIATION FOR
ORIENTATION SOUTH)
 * Wall Radiation:Total radiation on surface 2 ->    6- ITEAST  (INCIDENT RADIATION FOR
ORIENTATION EAST)
 * Wall Radiation:Total radiation on surface 4 ->    7- ITWEST  (INCIDENT RADIATION FOR
ORIENTATION WEST)
 * Horizontal Radiation:Total radiation on surface 1 ->    8- ITHORIZONT  (INCIDENT RADIATION
FOR ORIENTATION HORIZONT)
 * Wall Radiation:Beam radiation on surface 1 ->    9- IBNORTH  (INCIDENT BEAM RADIATION
FOR ORIENTATION NORTH)
 * Wall Radiation:Beam radiation on surface 3 ->   10- IBSOUTH  (INCIDENT BEAM RADIATION
FOR ORIENTATION SOUTH)
 * Wall Radiation:Beam radiation on surface 2 ->   11- IBEAST  (INCIDENT BEAM RADIATION FOR
ORIENTATION EAST)
 * Wall Radiation:Beam radiation on surface 4 ->   12- IBWEST  (INCIDENT BEAM RADIATION FOR
ORIENTATION WEST)
 * Horizontal Radiation:Beam radiation on surface 1 ->   13- IBHORIZONT  (INCIDENT BEAM
RADIATION FOR ORIENTATION HORIZONT)
 * Wall Radiation:Incidence angle for surface 1 ->   14- AINORTH  (ANGLE OF INCIDENCE FOR
ORIENTATION NORTH)
 * Wall Radiation:Incidence angle of surface 3 ->   15- AISOUTH  (ANGLE OF INCIDENCE FOR
ORIENTATION SOUTH)
 * Wall Radiation:Incidence angle of surface 2  ->   16- AIEAST  (ANGLE OF INCIDENCE FOR
ORIENTATION EAST)
 * Wall Radiation:Incidence angle of surface 4 ->   17- AIWEST  (ANGLE OF INCIDENCE FOR
ORIENTATION WEST)
 * Horizontal Radiation:Incidence angle for surface 1 ->   18- AIHORIZONT  (ANGLE OF INCIDENCE
FOR ORIENTATION HORIZONT)
 * Supply Flowrate:Qlobby ->   19- LOBBYVENT  (INPUT)
 * Supply Flowrate:Qwash ->   20- WASHVENT  (INPUT)
 * Supply Flowrate:Qoffice ->   21- OFFICEVENT  (INPUT)
 * Supply Flowrate:Qelev ->   22- ELEVATVENT  (INPUT)
 * Supply Flowrate:Qstairs ->   23- STAIRSVENT  (INPUT)
 * Density:Dry bulb temperature ->   24- TEMP  (INPUT)
 * Density:Percent relative humidity  ->   25- RELHUM  (INPUT)
      2,07         2,06         5,01         3,07         3,17
      3,12         3,22         4,07         3,08         3,18
      3,13         3,23         4,08         3,10         3,20
      3,15         3,25         4,10         QLOBBY       QWASH
      QOFFICE      QELEV        QSTAIRS      33,07        33,06
 *** INITIAL INPUT VALUES
      0.000E+00    0.000E+00    0.000E+00    0.000E+00    0.000E+00
      0.000E+00    0.000E+00    0.000E+00    0.000E+00    0.000E+00
      0.000E+00    0.000E+00    0.000E+00    0.000E+00    0.000E+00
      0.000E+00    0.000E+00    0.000E+00    0.000E+00    0.000E+00
```

```
      0.000E+00    0.000E+00    0.000E+00    0.000E+00    0.000E+00
 *** External files
ASSIGN C:\Thesiswork\Buildings\CV\Offices\Buildings\Office.bld        26
 *|? Which file contains the building description (*.BLD)? |1000
  ASSIGN C:\Thesiswork\Buildings\CV\Offices\Buildings\Office.trn        27
 *|? Which file contains the transfer function iformation (*.TRN)? |1000
  ASSIGN C:\trnsys15\prebid\Lib\American\W4-lib.dat                  28
 *|? Which file contains the window library for TYPE 56? |1000
  * Model "Exhaust Mixer" (Type 648)
   UNIT  36     TYPE 648     Exhaust Mixer
 *$UNIT_NAME Exhaust Mixer
 *$MODEL .\TESS Libraries\Hvac\Hydronics\Valves\100-Port Flow Mixer\Air\Humidity
Ratio\Type648.tmf
 *$POSITION 1995 199
 *$LAYER Main
      PARAMETERS   1
 * 1 Number of inlets
      5.000E+00
      INPUTS   15
 * Office:    1- (air temperature of zone)  TAIR   1 ->Air temperature at inlet-1
 * Office:   17- (absolute air humidity)  ABSHUM 1 ->Air humidity ratio at inlet-1
 * Supply:Flow rate at outlet -1 ->Dry air flow rate at inlet-1
 * Office:    2- (air temperature of zone)  TAIR   2 ->Air temperature at inlet-2
 * Office:   18- (absolute air humidity)  ABSHUM 2 ->Air humidity ratio at inlet-2
 * Supply:Flow rate at outlet -2 ->Dry air flow rate at inlet-2
 * Office:    3- (air temperature of zone)  TAIR   3 ->Air temperature at inlet-3
 * Office:   19- (absolute air humidity)  ABSHUM 3 ->Air humidity ratio at inlet-3
 * Supply:Flow rate at outlet -3 ->Dry air flow rate at inlet-3
 * Office:    5- (air temperature of zone)  TAIR   5 ->Air temperature at inlet-4
 * Office:   20- (absolute air humidity)  ABSHUM 5 ->Air humidity ratio at inlet-4
 * Supply:Flow rate at outlet -4 ->Dry air flow rate at inlet-4
 * Office:    6- (air temperature of zone)  TAIR   6 ->Air temperature at inlet-5
 * Office:   21- (absolute air humidity)  ABSHUM 6 ->Air humidity ratio at inlet-5
 * Supply:Flow rate at outlet -5 ->Dry air flow rate at inlet-5
      35,01          35,17          32,03          35,02          35,18
      32,06          35,03          35,19          32,09          35,05
      35,20          32,12          35,06          35,21          32,15
 *** INITIAL INPUT VALUES
      2.000E+01    8.000E-03    1.000E+03    2.000E+01    8.000E-03
      1.000E+03    2.000E+01    8.000E-03    1.000E+03    2.000E+01
      8.000E-03    1.000E+03    2.000E+01    8.000E-03    1.000E+03
  * Model "Exhaust Fan" (Type 662)
   UNIT  37     TYPE 662     Exhaust Fan
 *$UNIT_NAME Exhaust Fan
 *$MODEL .\TESS Libraries\Hvac\Hydronics\Fans\Variable Speed Fan\Type662.tmf
 *$POSITION 2096 295
 *$LAYER Main
 *$# NOTE: This component sets the flow rate for the rest of the components in the flow loop by
multiplying the
 *$# nominal flow rate (Parameter 1) by the  control signal (Input 3).  The mass flow rate input of this
component
 *$# is only for visualization purposes; it is not used except for convergence checking.
      PARAMETERS   6
 * 1 Rated air flow rate
 * 2 Specific heat of air
 * 3 Rated fan power
```

```
 * 4 Thermal conversion coefficient
 * 5 Number of power coefficients
 * 6 Power coefficient
      3.422E+05    1.007E+00    2.037E+05    3.000E-01    1.000E+00
      1.000E+00
      INPUTS    4
 * Exhaust Mixer:Air temperature at outlet  ->Inlet air temperature
 * Exhaust Mixer:Air humidity ratio at outlet  ->Inlet air humidity ratio
 * Exhaust Mixer:Dry air flow rate at outlet  ->Inlet mass flow rate
 * Supply Control:SupplyCont ->Control signal
      36,01         36,02         36,03         SUPPLYCONT
 *** INITIAL INPUT VALUES
      2.000E+01    8.000E-03    3.422E+05    1.000E+00
  * Model "Exhaust diverter" (Type 646)
   UNIT  38     TYPE 646     Exhaust diverter
 *$UNIT_NAME Exhaust diverter
 *$MODEL .\TESS Libraries\Hvac\Hydronics\Valves\100-Port Flow Diverter\Air\Humidity
Ratio\Type646.tmf
 *$POSITION 2193 360
 *$LAYER Main
      PARAMETERS   1
 * 1 Number of outlet ports
      2.000E+00
      INPUTS    5
 * Exhaust Fan:Outlet air temperature ->Inlet air temperature
 * Exhaust Fan:Outlet air humidity ratio ->Inlet air humidity ratio
 * Exhaust Fan:Outlet flow rate ->Inlet air flow rate
 * Es/El/X:X ->Fraction of flow to outlet -1
 * [unconnected] Fraction of flow to outlet -2
      37,01         37,02         37,03         X            CONST
 *** INITIAL INPUT VALUES
      2.000E+01    8.000E-03    3.422E+05    8.000E-01    2.000E-01
  * Model "Plotter" (Type 65)
   UNIT  39     TYPE  65     Plotter
 *$UNIT_NAME Plotter
 *$MODEL .\Output\Online Plotter\TYPE65.tmf
 *$POSITION 1965 -1
 *$LAYER Outputs
      PARAMETERS  10
 * 1 Nb. of left-axis variables
 * 2 Nb. of right-axis variables
 * 3 Left axis minimum
 * 4 Left axis maximum
 * 5 Right axis minimum
 * 6 Right axis maximum
 * 7 Number of plots per simulation
 * 8 X-axis gridpoints
 * 9 Shut off Online w/o removing
 * 10 Logical Unit for ouput file
      1.000E+00    1.000E+00    0.000E+00    4.000E+01    0.000E+00
      1.000E+02    1.000E+00    7.000E+00    0.000E+00   -1.000E+00
      INPUTS    2
 * Office:    3- (air temperature of zone)  TAIR   3 ->Left axis variable
 * Office:   11- (relativ humidity of zone...)  RELHUM 3 ->Right axis variable
      35,03         35,11
 *** INITIAL INPUT VALUES
```

```
      Office              Offi     ce
      LABELS  5
      C           %
      Temperature
      Relative Humidity
      Result
* Model "Int-temp" (Type 93)
  UNIT  40     TYPE  93     Int-temp
 *$UNIT_NAME Int-temp
 *$MODEL .\Utility\Input Value Recall\Type93.tmf
 *$POSITION 1628 36
 *$LAYER Main
 *$# The input value recal component remembers the values NI inputs for the preceeding NT timesteps
 *$# where NI and NT are set as parameters.
      PARAMETERS   2
 * 1 Number of inputs to be stored
 * 2 Number of timesteps to be stored
      8.000E+00    2.500E+01
      INPUTS    8
 * Office:    1- (air temperature of zone)  TAIR   1 ->Input value-1
 * Office:    2- (air temperature of zone)  TAIR   2 ->Input value-2
 * Office:    3- (air temperature of zone)  TAIR   3 ->Input value-3
 * Office:    4- (air temperature of zone)  TAIR   4 ->Input value-4
 * Office:    5- (air temperature of zone)  TAIR   5 ->Input value-5
 * Office:    6- (air temperature of zone)  TAIR   6 ->Input value-6
 * Office:    7- (air temperature of zone)  TAIR   7 ->Input value-7
 * Office:    8- (air temperature of zone)  TAIR   8 ->Input value-8
      35,01         35,02         35,03         35,04         35,05
      35,06         35,07         35,08
 *** INITIAL INPUT VALUES
      0.000E+00    0.000E+00    0.000E+00    0.000E+00    0.000E+00
      0.000E+00    0.000E+00    0.000E+00
  * EQUATIONS "MeanT-Lobby"
  EQUATIONS   2
    Sum1 =
[40,1]+[40,2]+[40,3]+[40,4]+[40,5]+[40,6]+[40,7]+[40,8]+[40,9]+[40,10]+[40,11]+[40,12]+[40,13]
    Sum2 =
(Sum1+[40,14]+[40,15]+[40,16]+[40,17]+[40,18]+[40,19]+[40,20]+[40,21]+[40,22]+[40,23]+[40,24]+[4
0,25])/25
 *$UNIT_NAME MeanT-Lobby
 *$LAYER Main
 *$POSITION 1826 36
  * EQUATIONS "MeanT-Wash"
  EQUATIONS   2
    Sum3 =
[40,26]+[40,27]+[40,28]+[40,29]+[40,30]+[40,31]+[40,32]+[40,33]+[40,34]+[40,35]+[40,36]+[40,37]+[4
0,38]
    Sum4 =
(Sum3+[40,39]+[40,40]+[40,41]+[40,42]+[40,43]+[40,44]+[40,45]+[40,46]+[40,47]+[40,48]+[40,49]+[4
0,50])/25
 *$UNIT_NAME MeanT-Wash
 *$LAYER Main
 *$POSITION 1835 100
  * EQUATIONS "MeanT-Office"
  EQUATIONS   2
```

```
    Sum5 =
[40,51]+[40,52]+[40,53]+[40,54]+[40,55]+[40,56]+[40,57]+[40,58]+[40,59]+[40,60]+[40,61]+[40,62]+[4
0,63]
    Sum6 =
(Sum5+[40,64]+[40,65]+[40,66]+[40,67]+[40,68]+[40,69]+[40,70]+[40,71]+[40,72]+[40,73]+[40,74]+[4
0,75])/25
 *$UNIT_NAME MeanT-Office
 *$LAYER Main
 *$POSITION 1835 154
  * EQUATIONS "MeanT-Plenum"
  EQUATIONS   2
    Sum13 =
[40,76]+[40,77]+[40,78]+[40,79]+[40,80]+[40,81]+[40,82]+[40,83]+[40,84]+[40,85]+[40,86]+[40,87]+[4
0,88]
    Sum14 =
(Sum13+[40,89]+[40,90]+[40,91]+[40,92]+[40,93]+[40,94]+[40,95]+[40,96]+[40,97]+[40,98]+[40,99]+[
40,100])/25
 *$UNIT_NAME MeanT-Plenum
 *$LAYER Main
 *$POSITION 1834 207
  * EQUATIONS "MeanT-Elevator"
  EQUATIONS   2
    Sum9 =
[40,101]+[40,102]+[40,103]+[40,104]+[40,105]+[40,106]+[40,107]+[40,108]+[40,109]+[40,110]+[40,11
1]+[40,112]+[40,113]
    Sum10 =
(Sum9+[40,114]+[40,115]+[40,116]+[40,117]+[40,118]+[40,119]+[40,120]+[40,121]+[40,122]+[40,123]
+[40,124]+[40,125])/25
 *$UNIT_NAME MeanT-Elevator
 *$LAYER Main
 *$POSITION 1841 271
  * EQUATIONS "MeanT-Stairs"
  EQUATIONS   2
    Sum11 =
[40,126]+[40,127]+[40,128]+[40,129]+[40,130]+[40,131]+[40,132]+[40,133]+[40,134]+[40,135]+[40,13
6]+[40,137]+[40,138]
    Sum12 =
(Sum11+[40,139]+[40,140]+[40,141]+[40,142]+[40,143]+[40,144]+[40,145]+[40,146]+[40,147]+[40,148
]+[40,149]+[40,150])/25
 *$UNIT_NAME MeanT-Stairs
 *$LAYER Main
 *$POSITION 1845 346
  * EQUATIONS "MeanT-Duct"
  EQUATIONS   2
    Sum15 =
[40,151]+[40,152]+[40,153]+[40,154]+[40,155]+[40,156]+[40,157]+[40,158]+[40,159]+[40,160]+[40,16
1]+[40,162]+[40,163]
    Sum16 =
(Sum15+[40,164]+[40,165]+[40,166]+[40,167]+[40,168]+[40,169]+[40,170]+[40,171]+[40,172]+[40,173
]+[40,174]+[40,175])/25
 *$UNIT_NAME MeanT-Duct
 *$LAYER Main
 *$POSITION 1855 420
  * EQUATIONS "MeanT-Mechanical"
  EQUATIONS   2
```

```
    Sum17 =
[40,176]+[40,177]+[40,178]+[40,179]+[40,180]+[40,181]+[40,182]+[40,183]+[40,184]+[40,185]+[40,18
6]+[40,187]+[40,188]
    Sum18 =
(Sum17+[40,189]+[40,190]+[40,191]+[40,192]+[40,193]+[40,194]+[40,195]+[40,196]+[40,197]+[40,198
]+[40,199]+[40,200])/25
*$UNIT_NAME MeanT-Mechanical
*$LAYER Main
*$POSITION 1855 484
 * Model "Temperature" (Type 25)
  UNIT  49    TYPE  25    Temperature
*$UNIT_NAME Temperature
*$MODEL .\Output\Printer\Don't Print Units to File\TYPE25c.tmf
*$POSITION 2225 62
*$LAYER Main
     PARAMETERS   4
* 1 Printing interval
* 2 Start time
* 3 Stop time
* 4 Logical unit
     1.000E+00    8.760E+03    1.752E+04    1.700E+01
     INPUTS    8
* MeanT-Lobby:Sum2 ->Input to be printed-1
* MeanT-Wash:Sum4 ->Input to be printed-2
* MeanT-Office:Sum6 ->Input to be printed-3
* MeanT-Plenum:Sum14 ->Input to be printed-4
* MeanT-Elevator:Sum10 ->Input to be printed-5
* MeanT-Stairs:Sum12 ->Input to be printed-6
* MeanT-Duct:Sum16 ->Input to be printed-7
* MeanT-Mechanical:Sum18 ->Input to be printed-8
     SUM2          SUM4          SUM6          SUM14         SUM10
     SUM12         SUM16         SUM18
*** INITIAL INPUT VALUES
     Lobby         Wash          Office        Plenum        Elev
     Stairs        Duct          Mech
*** External files
 ASSIGN C:\Thesiswork\Buildings\CV\Offices\Saskatoon\Results_saskato   17
*|? Which file should contain the printed results? |1000
* Model "Int-RH" (Type 93)
  UNIT  50    TYPE  93    Int-RH
*$UNIT_NAME Int-RH
*$MODEL .\Utility\Input Value Recall\Type93.tmf
*$POSITION 2295 143
*$LAYER Main
*$# The input value recal component remembers the values NI inputs for the preceeding NT timesteps
*$# where NI and NT are set as parameters.
     PARAMETERS   2
* 1 Number of inputs to be stored
* 2 Number of timesteps to be stored
     9.000E+00    2.500E+01
     INPUTS    9
* Office:    9- (relativ humidity of zone...)  RELHUM 1 ->Input value-1
* Office:   10- (relativ humidity of zone...)  RELHUM 2 ->Input value-2
* Office:   11- (relativ humidity of zone...)  RELHUM 3 ->Input value-3
* Office:   12- (relativ humidity of zone...)  RELHUM 4 ->Input value-4
* Office:   13- (relativ humidity of zone...)  RELHUM 5 ->Input value-5
```

```
 * Office:   14- (relativ humidity of zone...)  RELHUM 6 ->Input value-6
 * Office:   15- (relativ humidity of zone...)  RELHUM 7 ->Input value-7
 * Office:   16- (relativ humidity of zone...)  RELHUM 8 ->Input value-8
 * Office:   19- (absolute air humidity)  ABSHUM 3 ->Input value-9
      35,09          35,10          35,11          35,12          35,13
      35,14          35,15          35,16          35,19
 *** INITIAL INPUT VALUES
      0.000E+00    0.000E+00    0.000E+00    0.000E+00    0.000E+00
      0.000E+00    0.000E+00    0.000E+00    0.000E+00
  * EQUATIONS "MeanRH-Lobby"
   EQUATIONS   2
     Add1 =
[50,1]+[50,2]+[50,3]+[50,4]+[50,5]+[50,6]+[50,7]+[50,8]+[50,9]+[50,10]+[50,11]+[50,12]+[50,13]
     Add2 =
(Add1+[50,14]+[50,15]+[50,16]+[50,17]+[50,18]+[50,19]+[50,20]+[50,21]+[50,22]+[50,23]+[50,24]+[5
0,25])/25
 *$UNIT_NAME MeanRH-Lobby
 *$LAYER Main
 *$POSITION 2454 68
  * EQUATIONS "MeanRH-Wash"
   EQUATIONS   2
     Add3 =
[50,26]+[50,27]+[50,28]+[50,29]+[50,30]+[50,31]+[50,32]+[50,33]+[50,34]+[50,35]+[50,36]+[50,37]+[5
0,38]
     Add4 =
(Add3+[50,39]+[50,40]+[50,41]+[50,42]+[50,43]+[50,44]+[50,45]+[50,46]+[50,47]+[50,48]+[50,49]+[5
0,50])/25
 *$UNIT_NAME MeanRH-Wash
 *$LAYER Main
 *$POSITION 2453 132
  * EQUATIONS "MeanRH-Office"
   EQUATIONS   2
     Add5 =
[50,51]+[50,52]+[50,53]+[50,54]+[50,55]+[50,56]+[50,57]+[50,58]+[50,59]+[50,60]+[50,61]+[50,62]+[5
0,63]
     Add6 =
(Add5+[50,64]+[50,65]+[50,66]+[50,67]+[50,68]+[50,69]+[50,70]+[50,71]+[50,72]+[50,73]+[50,74]+[5
0,75])/25
 *$UNIT_NAME MeanRH-Office
 *$LAYER Main
 *$POSITION 2453 185
  * EQUATIONS "MeanRH-Plenum"
  EQUATIONS   2
     Add13 =
[50,76]+[50,77]+[50,78]+[50,79]+[50,80]+[50,81]+[50,82]+[50,83]+[50,84]+[50,85]+[50,86]+[50,87]+[5
0,88]
     Add14 =
(Add13+[50,89]+[50,90]+[50,91]+[50,92]+[50,93]+[50,94]+[50,95]+[50,96]+[50,97]+[50,98]+[50,99]+[
50,100])/25
 *$UNIT_NAME MeanRH-Plenum
 *$LAYER Main
 *$POSITION 2456 239
  * EQUATIONS "MeanRH-Elevator"
   EQUATIONS   2
```

```
    Add9 =
[50,101]+[50,102]+[50,103]+[50,104]+[50,105]+[50,106]+[50,107]+[50,108]+[50,109]+[50,110]+[50,11
1]+[50,112]+[50,113]
    Add10 =
(Add9+[50,114]+[50,115]+[50,116]+[50,117]+[50,118]+[50,119]+[50,120]+[50,121]+[50,122]+[50,123]
+[50,124]+[50,125])/25
 *$UNIT_NAME MeanRH-Elevator
 *$LAYER Main
 *$POSITION 2459 292
  * EQUATIONS "MeanRH-Stairs"
  EQUATIONS   2
    Add11 =
[50,126]+[50,127]+[50,128]+[50,129]+[50,130]+[50,131]+[50,132]+[50,133]+[50,134]+[50,135]+[50,13
6]+[50,137]+[50,138]
    Add12 =
(Add11+[50,139]+[50,140]+[50,141]+[50,142]+[50,143]+[50,144]+[50,145]+[50,146]+[50,147]+[50,148
]+[50,149]+[50,150])/25
 *$UNIT_NAME MeanRH-Stairs
 *$LAYER Main
 *$POSITION 2463 345
  * EQUATIONS "MeanRH-Duct"
   EQUATIONS   2
    Add15 =
[50,151]+[50,152]+[50,153]+[50,154]+[50,155]+[50,156]+[50,157]+[50,158]+[50,159]+[50,160]+[50,16
1]+[50,162]+[50,163]
    Add16 =
(Add15+[50,164]+[50,165]+[50,166]+[50,167]+[50,168]+[50,169]+[50,170]+[50,171]+[50,172]+[50,173
]+[50,174]+[50,175])/25
 *$UNIT_NAME MeanRH-Duct
 *$LAYER Main
 *$POSITION 2472 399
  * EQUATIONS "MeanRH-Mechanical"
   EQUATIONS   2
    Add17 =
[50,176]+[50,177]+[50,178]+[50,179]+[50,180]+[50,181]+[50,182]+[50,183]+[50,184]+[50,185]+[50,18
6]+[50,187]+[50,188]
    Add18 =
(Add17+[50,189]+[50,190]+[50,191]+[50,192]+[50,193]+[50,194]+[50,195]+[50,196]+[50,197]+[50,198
]+[50,199]+[50,200])/25
 *$UNIT_NAME MeanRH-Mechanical
 *$LAYER Main
 *$POSITION 2468 452
  * EQUATIONS "MeanW"
   EQUATIONS   2
    Add25 =
[50,201]+[50,202]+[50,203]+[50,204]+[50,205]+[50,206]+[50,207]+[50,208]+[50,209]+[50,210]+[50,21
1]+[50,212]+[50,213]
    Add26 =
(Add25+[50,214]+[50,215]+[50,216]+[50,217]+[50,218]+[50,219]+[50,220]+[50,221]+[50,222]+[50,223
]+[50,224]+[50,225])/25
 *$UNIT_NAME MeanW
 *$LAYER Main
 *$POSITION 2469 516
  * Model "Relative Humidity" (Type 25)
  UNIT  60     TYPE  25     Relative Humidity
 *$UNIT_NAME Relative Humidity
```

```
*$MODEL .\Output\Printer\Don't Print Units to File\TYPE25c.tmf
*$POSITION 2217 136
*$LAYER Main
     PARAMETERS   4
* 1 Printing interval
* 2 Start time
* 3 Stop time
* 4 Logical unit
     1.000E+00    8.760E+03    1.752E+04    1.800E+01
     INPUTS    9
* MeanRH-Lobby:Add2 ->Input to be printed-1
* MeanRH-Wash:Add4 ->Input to be printed-2
* MeanRH-Office:Add6 ->Input to be printed-3
* MeanRH-Plenum:Add14 ->Input to be printed-4
* MeanRH-Elevator:Add10 ->Input to be printed-5
* MeanRH-Stairs:Add12 ->Input to be printed-6
* MeanRH-Duct:Add16 ->Input to be printed-7
* MeanRH-Mechanical:Add18 ->Input to be printed-8
* MeanW:Add26 ->Input to be printed-9
     ADD2          ADD4          ADD6          ADD14         ADD10
     ADD12         ADD16         ADD18         ADD26
*** INITIAL INPUT VALUES
     Lobby         Wash          Office        Plen          Elev
     Stairs        Duct          Mech          Win
*** External files
 ASSIGN C:\Thesiswork\Buildings\CV\Offices\Saskatoon\Results_saskato   18
*|? Which file should contain the printed results? |1000
  END
       TRANSIENT SIMULATION     STARTING AT TIME =  8.737E+03
                         STOPPING AT TIME =  1.752E+04
                                TIMESTEP =     1/25
          DIFFERENTIAL EQUATION ERROR TOLERANCE =  1.000E-03
               ALGEBRAIC CONVERGENCE TOLERANCE =  1.000E-01
1  TRNSYS COMPONENT OUTPUT MAP
 UNIT  1 TYPE 89  UNIT/TYPE/INPUT
     OUTPUT  5       2 33  1
     OUTPUT  9       2 33  2
     OUTPUT  4       3 16  1
     OUTPUT  3       3 16  2
     OUTPUT 99       3 16  3
     OUTPUT100       3 16  4
     OUTPUT  4       4 16  1
     OUTPUT  3       4 16  2
     OUTPUT 99       4 16  3
     OUTPUT100       4 16  4
     OUTPUT  5       5 69  1
     OUTPUT  9       5 69  2
 UNIT  2 TYPE 33  UNIT/TYPE/INPUT
     OUTPUT  7      18667  4
     OUTPUT  6      18667  5
     OUTPUT  7      24651  1
                   35 56  1
     OUTPUT  6      35 56  2
 UNIT  3 TYPE 16  UNIT/TYPE/INPUT
     OUTPUT  7      35 56  4
     OUTPUT 17      35 56  5
```

```
    OUTPUT 12      35 56  6
    OUTPUT 22      35 56  7
    OUTPUT  8      35 56  9
    OUTPUT 18      35 56 10
    OUTPUT 13      35 56 11
    OUTPUT 23      35 56 12
    OUTPUT 10      35 56 14
    OUTPUT 20      35 56 15
    OUTPUT 15      35 56 16
    OUTPUT 25      35 56 17
UNIT  4 TYPE 16  UNIT/TYPE/INPUT
    OUTPUT  5       5 69  3
    OUTPUT  6       5 69  4
    OUTPUT  7      35 56  8
    OUTPUT  8      35 56 13
    OUTPUT 10      35 56 18
UNIT  5 TYPE 69  UNIT/TYPE/INPUT
    OUTPUT  1      35 56  3
UNIT  6 TYPE 14  UNIT/TYPE/INPUT
    OUTPUT  2       7  8  3
UNIT  7 TYPE  8  UNIT/TYPE/INPUT
UNIT  8 TYPE 14  UNIT/TYPE/INPUT
UNIT 11 TYPE 33  UNIT/TYPE/INPUT
    OUTPUT  7      18667  1
    OUTPUT  6      18667  2
UNIT 18 TYPE667  UNIT/TYPE/INPUT
UNIT 21 TYPE648  UNIT/TYPE/INPUT
    OUTPUT  1      22 33  1
    OUTPUT  2      22 33  2
    OUTPUT  1      23643  1
    OUTPUT  2      23643  2
    OUTPUT  1      24651  2
    OUTPUT  2      24651  3
UNIT 22 TYPE 33  UNIT/TYPE/INPUT
UNIT 23 TYPE643  UNIT/TYPE/INPUT
UNIT 24 TYPE651  UNIT/TYPE/INPUT
    OUTPUT  1      26 33  1
                28 33  1
UNIT 26 TYPE 33  UNIT/TYPE/INPUT
UNIT 28 TYPE 33  UNIT/TYPE/INPUT
UNIT 31 TYPE662  UNIT/TYPE/INPUT
    OUTPUT  1      32646  1
    OUTPUT  2      32646  2
    OUTPUT  3      32646  3
UNIT 32 TYPE646  UNIT/TYPE/INPUT
    OUTPUT  1      33 33  1
    OUTPUT  2      33 33  2
    OUTPUT  3      36648  3
    OUTPUT  6      36648  6
    OUTPUT  9      36648  9
    OUTPUT 12      36648 12
    OUTPUT 15      36648 15
UNIT 33 TYPE 33  UNIT/TYPE/INPUT
    OUTPUT  7      35 56 24
    OUTPUT  6      35 56 25
UNIT 35 TYPE 56  UNIT/TYPE/INPUT
```

```
    OUTPUT  3       7  8  1
    OUTPUT  1      36648  1
    OUTPUT 17      36648  2
    OUTPUT  2      36648  4
    OUTPUT 18      36648  5
    OUTPUT  3      36648  7
    OUTPUT 19      36648  8
    OUTPUT  5      36648 10
    OUTPUT 20      36648 11
    OUTPUT  6      36648 13
    OUTPUT 21      36648 14
    OUTPUT  3      39 65  1
    OUTPUT 11      39 65  2
    OUTPUT  1      40 93  1
    OUTPUT  2      40 93  2
    OUTPUT  3      40 93  3
    OUTPUT  4      40 93  4
    OUTPUT  5      40 93  5
    OUTPUT  6      40 93  6
    OUTPUT  7      40 93  7
    OUTPUT  8      40 93  8
    OUTPUT  9      50 93  1
    OUTPUT 10      50 93  2
    OUTPUT 11      50 93  3
    OUTPUT 12      50 93  4
    OUTPUT 13      50 93  5
    OUTPUT 14      50 93  6
    OUTPUT 15      50 93  7
    OUTPUT 16      50 93  8
    OUTPUT 19      50 93  9
UNIT 36 TYPE648  UNIT/TYPE/INPUT
    OUTPUT  1      37662  1
    OUTPUT  2      37662  2
    OUTPUT  3      37662  3
UNIT 37 TYPE662  UNIT/TYPE/INPUT
    OUTPUT  1      38646  1
    OUTPUT  2      38646  2
    OUTPUT  3      38646  3
UNIT 38 TYPE646  UNIT/TYPE/INPUT
    OUTPUT  4      11 33  1
    OUTPUT  5      11 33  2
    OUTPUT  1      21648  4
    OUTPUT  2      21648  5
    OUTPUT  3      21648  6
UNIT 39 TYPE 65  UNIT/TYPE/INPUT
UNIT 40 TYPE 93  UNIT/TYPE/INPUT
UNIT 50 TYPE 93  UNIT/TYPE/INPUT
EQUATIONS        UNIT/TYPE/INPUT
    SUPPLYCONT         31662  4
                37662  4
    HCNEW              23643  4
    CCNEW              24651  5
    ES              18667  7
    EL              18667  8
    X               38646  4
    MOUT               18667  3
```

```
              18667  6
     EWCONTROL          18667  9
     MVENT             21648  3
     TVENT             21648  1
     WVENT             21648  2
     MMIXED            23643  3
              24651  4
     RHLT              26 33  2
     WCOOL             28 33  2
     TEMP              31662  1
     HUMIDITY          31662  2
     FLOWRATE          31662  3
     QLOBBY            35 56 19
     QWASH             35 56 20
     QOFFICE           35 56 21
     QELEV             35 56 22
     QSTAIRS           35 56 23
     SUM2              49 25  1
     SUM4              49 25  2
     SUM6              49 25  3
     SUM14             49 25  4
     SUM10             49 25  5
     SUM12             49 25  6
     SUM16             49 25  7
     SUM18             49 25  8
     ADD2              60 25  1
     ADD4              60 25  2
     ADD6              60 25  3
     ADD14             60 25  4
     ADD10             60 25  5
     ADD12             60 25  6
     ADD16             60 25  7
     ADD18             60 25  8
     ADD26             60 25  9
    DIFFERENTIAL EQUATIONS SOLVED BY MODIFIED EULER
 ***** NOTE *****
 UNIT   1 TYPE  89 DATA READER (EnergyPlus WEATHER FILE)
 LOCATION: Saskatoon              SK
 TIME ZONE:  6 HOURS W
 LATITUDE:    52.17 DEG  LONGITUDE:   106.68 DEG W
 ELEVATION:  504 M
 HOURS OF E+ DATA ARE IN LOCAL STANDARD TIME, NOT SOLAR TIME.
 BE SURE TO PERFORM RADIATION PROCESSING APPROPRIATELY.
 SHIFT IN SOLAR TIME HOUR ANGLE (L_STANDARD - L_LOCAL): -16.68 DEG
  20 STORAGE VARIABLES ALLOCATED FOR UNIT   3, TYPE  16 ENDING AT    21
  20 STORAGE VARIABLES ALLOCATED FOR UNIT   4, TYPE  16 ENDING AT    41
   5 STORAGE VARIABLES ALLOCATED FOR UNIT   5, TYPE  69 ENDING AT    46
   1 STORAGE VARIABLES ALLOCATED FOR UNIT  24, TYPE 651 ENDING AT    47
 TYPE56: ACTIVE LAYERS AND INTERNAL WINDOW CALCULATIONS INTEGRATED
!Evaporator air flow rates in liters per second (l/s)
 !Indoor air wet bulb temperatures (C)
 !Condenser temperatures (C)
    PERFORMANCE DATA FROM LOGICAL UNIT 11
  X3=   6.519E+04X2=   1.722E+01
     X1          Y1          Y2
   2.944E+01   1.727E+03   3.422E+05
```

```
  3.500E+01   1.660E+03   3.885E+05
  4.056E+01   1.600E+03   4.347E+05
  4.611E+01   1.531E+03   4.833E+05
 X3=   6.519E+04X2=   1.944E+01
     X1          Y1          Y2
  2.944E+01   1.822E+03   3.446E+05
  3.500E+01   1.762E+03   3.885E+05
  4.056E+01   1.695E+03   4.370E+05
  4.611E+01   1.626E+03   4.856E+05
 X3=   6.519E+04X2=   2.167E+01
     X1          Y1          Y2
  2.944E+01   1.924E+03   3.469E+05
  3.500E+01   1.864E+03   3.908E+05
  4.056E+01   1.790E+03   4.394E+05
  4.611E+01   1.720E+03   4.879E+05
 X3=   7.921E+04X2=   1.722E+01
     X1          Y1          Y2
  2.944E+01   1.811E+03   3.446E+05
  3.500E+01   1.741E+03   3.885E+05
  4.056E+01   1.674E+03   4.347E+05
  4.611E+01   1.593E+03   4.856E+05
 X3=   7.921E+04X2=   1.944E+01
     X1          Y1          Y2
  2.944E+01   1.905E+03   3.446E+05
  3.500E+01   1.836E+03   3.885E+05
  4.056E+01   1.769E+03   4.370E+05
  4.611E+01   1.695E+03   4.856E+05
 X3=   7.921E+04X2=   2.167E+01
     X1          Y1          Y2
  2.944E+01   2.007E+03   3.469E+05
  3.500E+01   1.938E+03   3.908E+05
  4.056E+01   1.871E+03   4.394E+05
  4.611E+01   1.797E+03   4.902E+05
 X3=   9.318E+04X2=   1.722E+01
     X1          Y1          Y2
  2.944E+01   1.871E+03   3.446E+05
  3.500E+01   1.797E+03   3.885E+05
  4.056E+01   1.727E+03   4.370E+05
  4.611E+01   1.653E+03   4.856E+05
 X3=   9.318E+04X2=   1.944E+01
     X1          Y1          Y2
  2.944E+01   1.966E+03   3.469E+05
  3.500E+01   1.898E+03   3.908E+05
  4.056E+01   1.822E+03   4.394E+05
  4.611E+01   1.748E+03   4.879E+05
 X3=   9.318E+04X2=   2.167E+01
     X1          Y1          Y2
  2.944E+01   2.060E+03   3.469E+05
  3.500E+01   1.993E+03   3.931E+05
  4.056E+01   1.917E+03   4.417E+05
  4.611E+01   1.843E+03   4.925E+05
! Indoor dry bulb temperature (C)
! Evaporator flow rate (l/s)
! Indoor wet bulb temperature (C)
! Sensible heat ratio (SHR) for 23.89/100321/17.22
! Sensible heat ratio (SHR) for 23.89/100321/19.44
```

! Sensible heat ratio (SHR) for 23.89/100321/21.67
! Sensible heat ratio (SHR) for 23.89/121820/17.22
! Sensible heat ratio (SHR) for 23.89/121820/19.44
! Sensible heat ratio (SHR) for 23.89/121820/21.67
! Sensible heat ratio (SHR) for 23.89/143316/17.22
! Sensible heat ratio (SHR) for 23.89/143316/19.44
! Sensible heat ratio (SHR) for 23.89/143316/21.67
! Sensible heat ratio (SHR) for 26.67/100321/17.22
! Sensible heat ratio (SHR) for 26.67/100321/19.44
! Sensible heat ratio (SHR) for 26.67/100321/21.67
! Sensible heat ratio (SHR) for 26.67/121820/17.22
! Sensible heat ratio (SHR) for 26.67/121820/19.44
! Sensible heat ratio (SHR) for 26.67/121820/21.67
! Sensible heat ratio (SHR) for 26.67/143316/17.22
! Sensible heat ratio (SHR) for 26.67/143316/19.44
! Sensible heat ratio (SHR) for 26.67/143316/21.67
! Sensible heat ratio (SHR) for 29.44/100321/17.22
! Sensible heat ratio (SHR) for 29.44/100321/19.44
! Sensible heat ratio (SHR) for 29.44/100321/21.67
! Sensible heat ratio (SHR) for 29.44/121820/17.22
! Sensible heat ratio (SHR) for 29.44/121820/19.44
! Sensible heat ratio (SHR) for 29.44/121820/21.67
! Sensible heat ratio (SHR) for 29.44/143316/17.22
! Sensible heat ratio (SHR) for 29.44/143316/19.44
! Sensible heat ratio (SHR) for 29.44/143316/21.67

```
   PERFORMANCE DATA FROM LOGICAL UNIT 12
 X3=   2.389E+01X2=   6.519E+04
     X1          Y1
  1.722E+01   6.900E-01
  1.944E+01   5.400E-01
  2.167E+01   4.100E-01
 X3=   2.389E+01X2=   7.921E+04
     X1          Y1
  1.722E+01   7.300E-01
  1.944E+01   5.700E-01
  2.167E+01   4.200E-01
 X3=   2.389E+01X2=   9.318E+04
     X1          Y1
  1.722E+01   7.700E-01
  1.944E+01   5.900E-01
  2.167E+01   4.300E-01
 X3=   2.667E+01X2=   6.519E+04
     X1          Y1
  1.722E+01   8.400E-01
  1.944E+01   6.800E-01
  2.167E+01   5.400E-01
 X3=   2.667E+01X2=   7.921E+04
     X1          Y1
  1.722E+01   9.000E-01
  1.944E+01   7.100E-01
  2.167E+01   5.600E-01
 X3=   2.667E+01X2=   9.318E+04
     X1          Y1
  1.722E+01   9.600E-01
  1.944E+01   7.700E-01
  2.167E+01   5.800E-01
```

```
X3=   2.944E+01X2=   6.519E+04
     X1          Y1
  1.722E+01   9.800E-01
  1.944E+01   8.200E-01
  2.167E+01   6.800E-01
X3=   2.944E+01X2=   7.921E+04
     X1          Y1
  1.722E+01   1.000E+00
  1.944E+01   8.800E-01
  2.167E+01   7.100E-01
X3=   2.944E+01X2=   9.318E+04
     X1          Y1
  1.722E+01   1.000E+00
  1.944E+01   9.400E-01
  2.167E+01   7.600E-01
                              HR : MIN :  SEC
         Start-Time:               11 :  58 : 21.54
           End-Time:               12 :  20 :  8.46
Total Execution Time:               0 :  21 : 46.92
                UNIT   1 WAS CALLED    219577 TIMES
                       2           219577
                       3           219577
                       4           219577
                       5           219577
                       6           219577
                       7           219577
                       8           219577
                      11            219657
                      18            220282
                      21            252153
                      22            232532
                      23             75811
                      24             77907
                      26            232608
                      28            232608
                      31            247348
                      32            229301
                      33            229301
                      35            229301
                      36            220860
                      37            219577
                      38            219586
                      39            219578
                      40            220860
                      49              8761
                      50            219607
                      60              8761
  *-------------------------------------------------------------*
   *    UNIT                  # ITERATIVE CALLS                 *
   *    1   -----------------         219577                    *
   *    2   -----------------         219577                    *
   *    3   -----------------         219577                    *
   *    4   -----------------         219577                    *
   *    5   -----------------         219577                    *
   *    6   -----------------         219577                    *
   *    7   -----------------         219577                    *
```

```
*    8   -----------------        219577                    *
*   11   -----------------        219657                    *
*   18   -----------------        220282                    *
*   21   --------------------     252153                    *
*   22   ------------------       232532                    *
*   23   ------                  75811                  *
*   24   ------                  77907                  *
*   26   ------------------       232608                    *
*   28   ------------------       232608                    *
*   31   -------------------      247348                    *
*   32   ------------------       229301                    *
*   33   ------------------       229301                    *
*   35   ------------------       229301                    *
*   36   -----------------        220860                    *
*   37   -----------------        219577                    *
*   38   -----------------        219586                    *
*   40   -----------------        220860                    *
*   50   -----------------        219607                    *
*                                                *
*  Total Iterative Component Calls =    5573015                *
*----------------------------------------------------------------*
```